AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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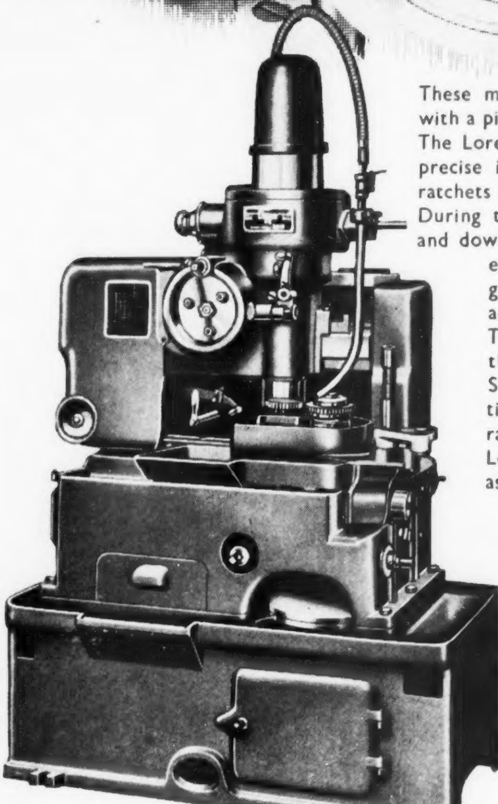
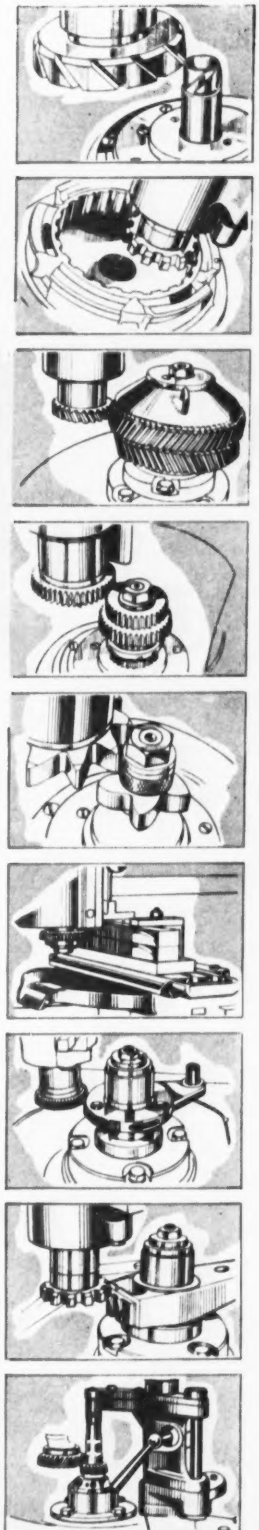
LORENZ

AUTOMATIC HIGH-SPEED GEAR SHAPING MACHINES





These machines shape by the generating process with a pinion type cutter for spur and helical gears. The Lorenz method is ideal for the production of precise internal and external gearing, sprockets, ratchets and other profiles.

During the cutting stroke the tool is moved up and down and rotated so that the tool and work execute the movement of two meshing gears. This method ensures maximum accuracy and smooth flanks of the work. The tool is re-sharpened by simply grinding the tooth face.

Special attachments are available for generating helical gears, gear trains, face gears, rack gears, shaping between centres, etc. Lorenz gear shapers are made in five sizes as shown below.

AUTOMATIC GEAR SHAPERS

MODEL	S 00	SV 00	S 5	S7/500	S7/1000
Pitch diam.	7⅛"	7⅛"	17¾"	19⅝"	40"
Face width	2"	2"	3"	5⅛"	6.5/16"
Diam. pitch/ module	6/4	6/4	5/5	3,6/7	3,6/7

AUTOMATIC GEAR HOBBERS

High production machines of extreme accuracy for the manufacture of spur, helical, worm wheels, and special gearings are made for work of pitch diameters from 23½" to 63" and diam. pitch/module 4.23/6 to 1.59/16.

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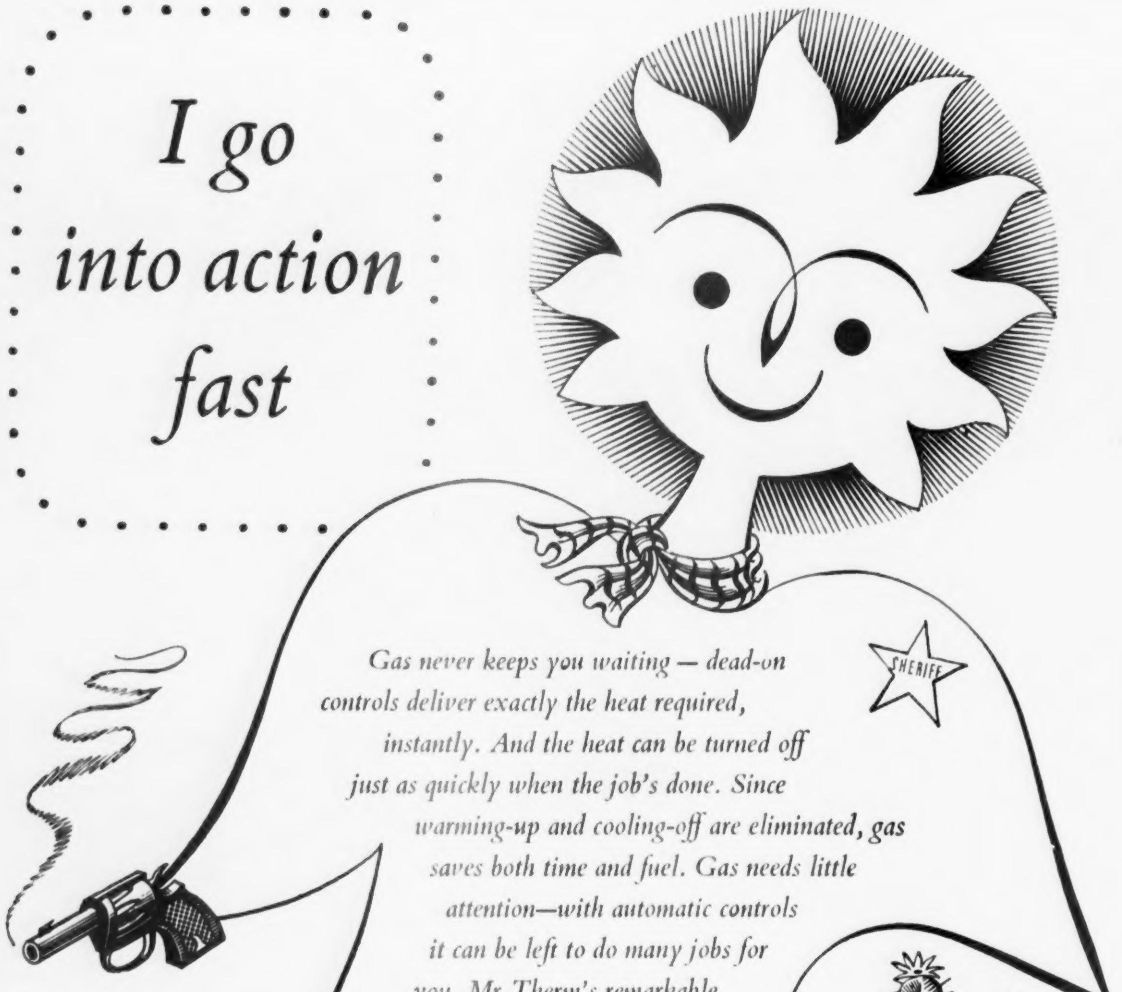
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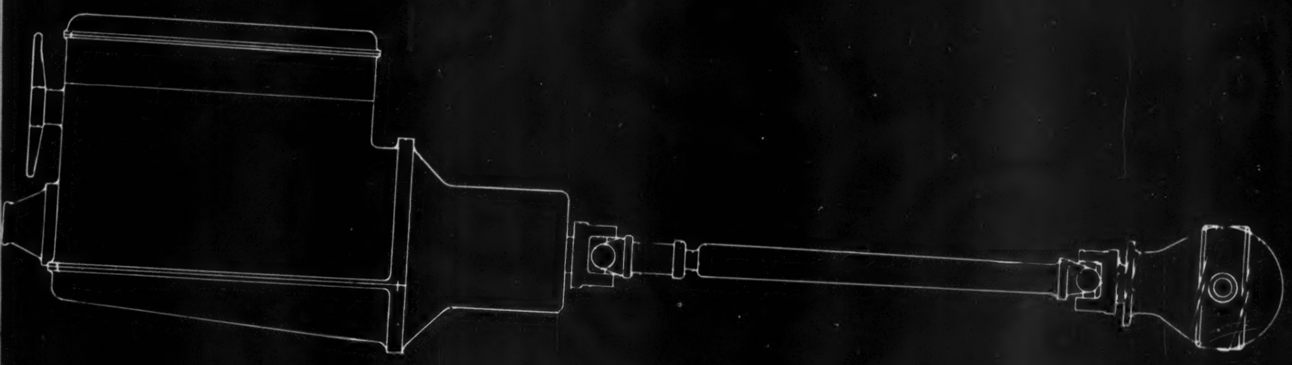


TRANSMISSION CONSIDERATIONS 2

JUDDER (I)

One example of bad manners in a transmission system is judder. The word, absent from many dictionaries, may well be the offspring of *jerk* and *shudder*, with *joggle* as god-parent. Be that as it may, judder denotes a jerky shudder occurring repetitively in the transmission at low speeds. Often it is sufficiently violent to cause some dismay to the occupants of the vehicle : always it is objectionable.

Basically, judder is caused by reversals of loan in the transmission, occurring repeatedly in some sort of cycle until a change of circumstance (e.g. speed or torque) kills them.

It can result from a faulty sparking plug or other defect causing exceptional fluctuations of torque. At low speeds, these fluctuations pass out of the control of the flywheel and can then cause reversals of load at the various clearances in the transmission. The result is a hammering or thumping at those places.

Judder can be caused also by the flexible supports of the power unit. Although these are intended primarily to allow motion of the engine about a longitudinal axis, they usually permit some movement parallel to that axis. This is undesirable, particularly when the back-axle torque reaction is resisted only by the road springs in what is often termed a Hotchkiss drive.

Consider the conditions in such a vehicle when starting from rest. As the clutch begins to engage, torque is applied to the driving axle and the reaction causes its casing to turn in a direction opposite to that of the road wheels. The springs resist this motion but, being flexible, they do not prevent it entirely. The nose of the axle therefore rises and may pull the propeller shaft backwards.

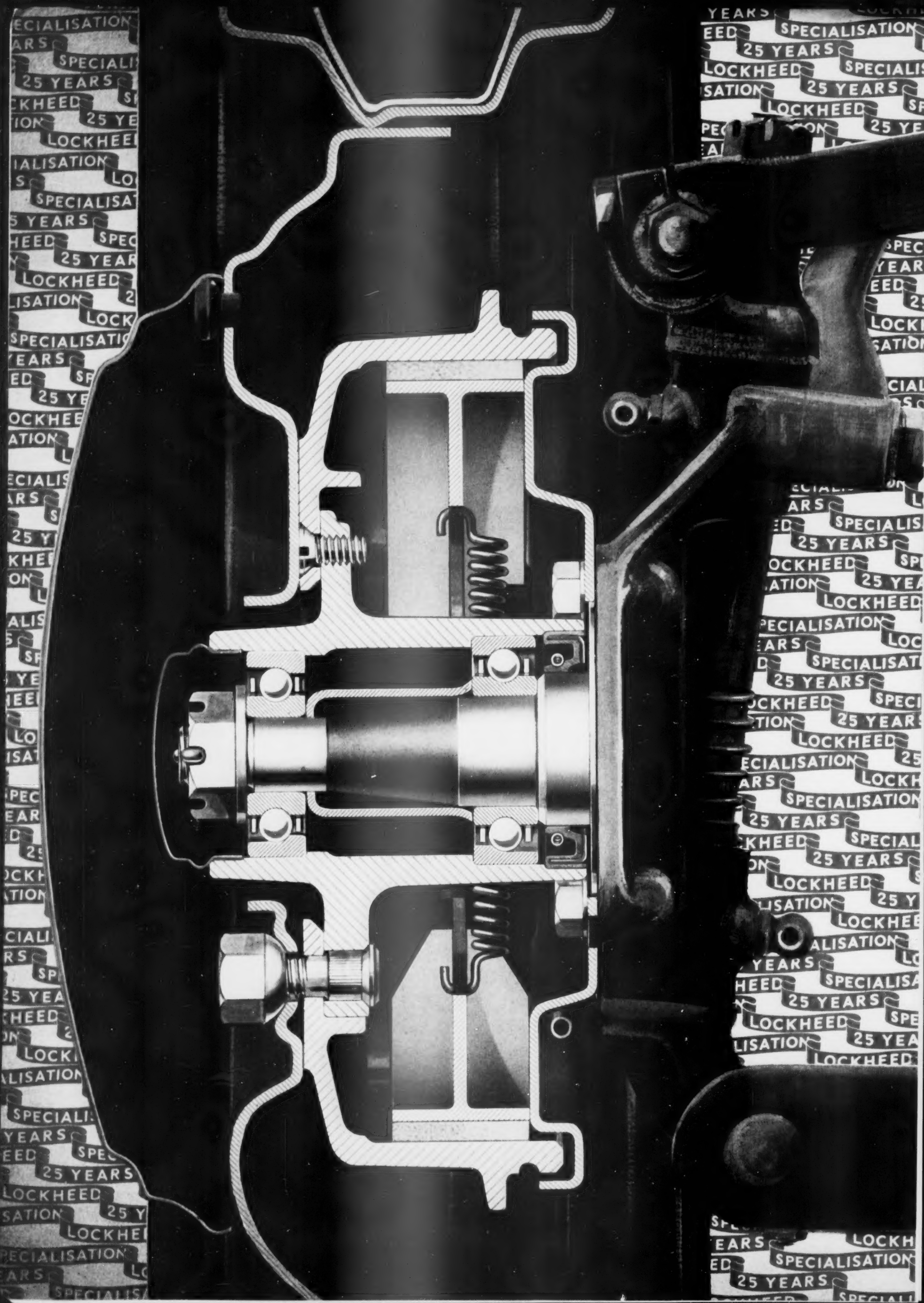
In passing, it may be noted that the axis of rotation of the casing is not necessarily the centre line of the axle shafts. It may be somewhere between that and the road springs. The case here considered is of a spiral bevel axle with underslung springs. With a hypoid axle placed beneath the springs, for example, the propeller shaft might be pushed forward.

A splined sliding joint in the propeller shaft takes care of excessive movement but in the present case the splines are already loaded. Consequently their friction resists sliding. So the power unit is moved with the propeller shaft. The amount of its movement will depend, of course, upon the longitudinal softness of its mounting system as a whole, as well as upon the angular movement of the axle casing.

Some consequences of this movement must depend upon the particular arrangement of the clutch withdrawal mechanism. If, as in the simplest case, the pedal is mounted on the clutch withdrawal shaft, events are easy to follow. The pedal pad being already against the driver's foot, rearward motion of the engine causes a partial disengagement of the clutch. This in turn reduces the torque applied to the axle and the latter therefore returns towards its normal static position. Consequently the engine moves forward again on its mountings and the clutch is re-engaged by a corresponding amount.

There will thus be repetitions of a cycle in which the clutch is alternately engaged and disengaged, with corresponding variations of torque. These are conditions which can cause judder. They occur also if the pedal is mounted on some part of the chassis frame and is connected to the clutch by a simple link. By a more complex linkage, this effect can be avoided ; and that can be done also by hydraulic operation of the clutch or by the use of some simple servo unit.

Judder, it should now be clear, is an annoying trouble, which shows itself in one place (the central region of the transmission) but originates in another (the extremities of the system). It can be reduced or even eliminated provided its origin is known and we shall return to this subject shortly.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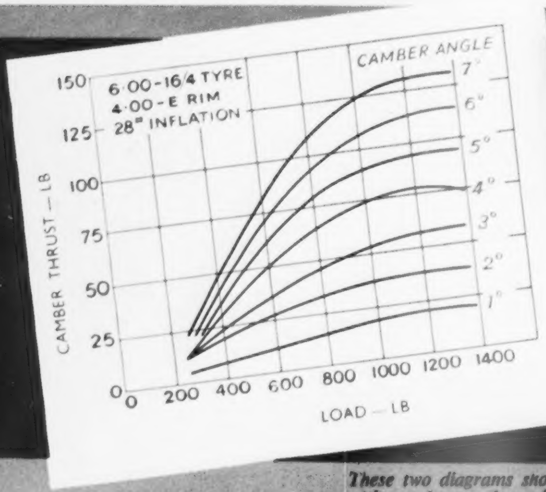
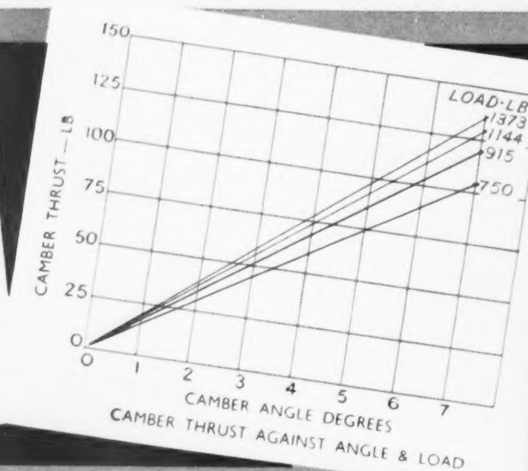
SUBTLETIES OF STEERING

Causes of over- and under-steer: 3. Tyre properties (b)





These two diagrams show how camber thrust varies with camber angle and load carried by the tyre.

WE have seen how the cornering power and self-centring torque developed by a rolling tyre subjected to a sideways load, and the drift angle which that sideways load produces, can be explained and understood in terms of the sideways flexibility of the different 'elements' which comprise the tread of that tyre. A further property of the rolling pneumatic tyre which can be explained and understood in the same way is camber thrust.

When a rolling tyre is leaned over sideways it tries to push itself in the direction in which it is leaning, as Mr. Wilson-Jones has said (Proc. I.Mech. E. Vol. 1). This property provides most of the cornering power required to persuade motor cycles to change their direction. This is understandable if we take the simpler way of explaining how camber thrust is developed. If a vertical tyre and wheel are deflected by a vertical load, and then leaned over to an angle ϕ with the vertical, the deflecting load W will now have components $W \cos \phi$ vertically and $W \sin \phi$ horizontally, and the ratio of sideways to vertical load is $\sin \phi \div \cos \phi$ or $\tan \phi$. Camber thrust is therefore $W \tan \phi$. In fact this answer is not a long way from the truth as observed experimentally. As Mr. Wilson-Jones has pointed out, if this were completely true, if the motor cycle and rider leaned over as one, and if the difference in angle between front and rear wheels due to the combination of steering angle and head tube rake were ignored, then a motor cycle would corner with no drift angles on the tyres.

The fourth feature of a rolling tyre and wheel subjected to a sideways load, and hence developing a sideways drift angle, is a very considerable increase in drag. If the sideways load is measured at right angles to the drifted path of the wheel and not perpendicular to its plane (which is correct because centrifugal force on a cornering car will act at right angles to the path of travel of the wheel) then a component of the actual sideways load at right angles to the wheel plane (which must be the true sideways load unless the brake is on) acts in a direction to increase the 'drag' of the tyre, by an amount $C \tan \theta$, where C is the cornering load and θ is the drift angle. The total drag is the sum of this and the drag or rolling resistance which would normally occur with a tyre not subjected to sideways load.

All four properties of tyres mentioned in this and the preceding essay — drift angle under cornering force, precessional or self-centring torque, camber thrust, and increased drag when drifting, were first studied and evaluated by running a loaded wheel and tyre on a drum. The difference between a drum surface and a flat road surface did to some extent affect the answers, but the variations caused by this difference have in most cases now been determined and the true answers are now known.

Tyre inflation, not previously mentioned, does affect the different properties. In effect it increases the rated load of a tyre, and thus shifts the whole family of cornering power — vertical load curves at different drift angles to the right.

This completes the brief survey of the properties of the pneumatic tyre which affect the cornering behaviour of the complete car. Before we can see the way in which each property makes its individual contribution to car behaviour, we must investigate the properties of the car which affect its cornering behaviour, and the way in which both tyre and car properties combine to determine car behaviour will then eventually become clear.

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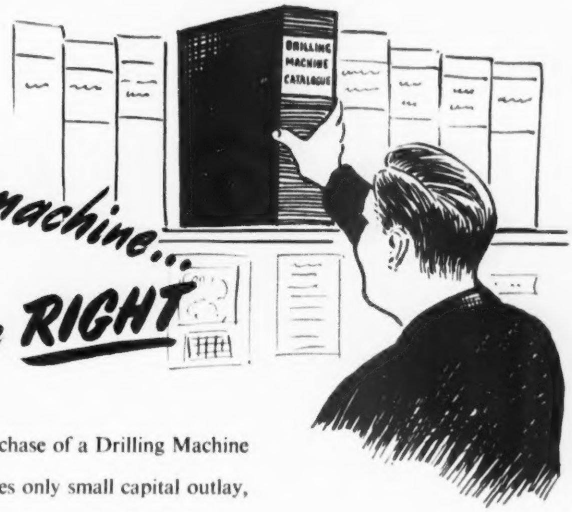

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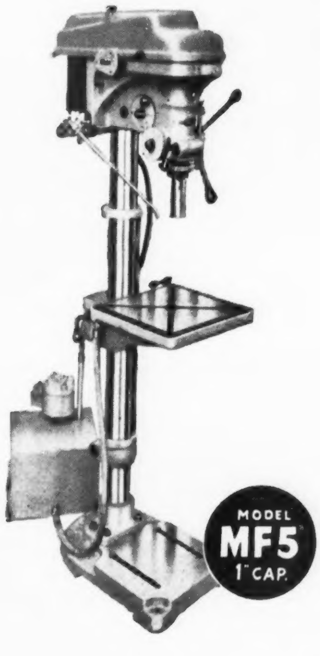

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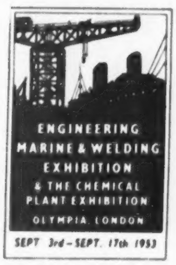

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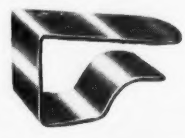

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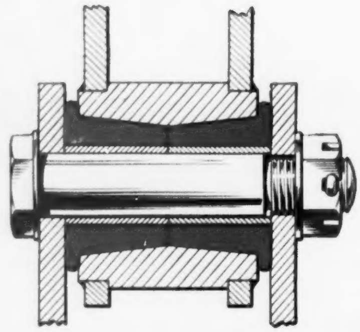

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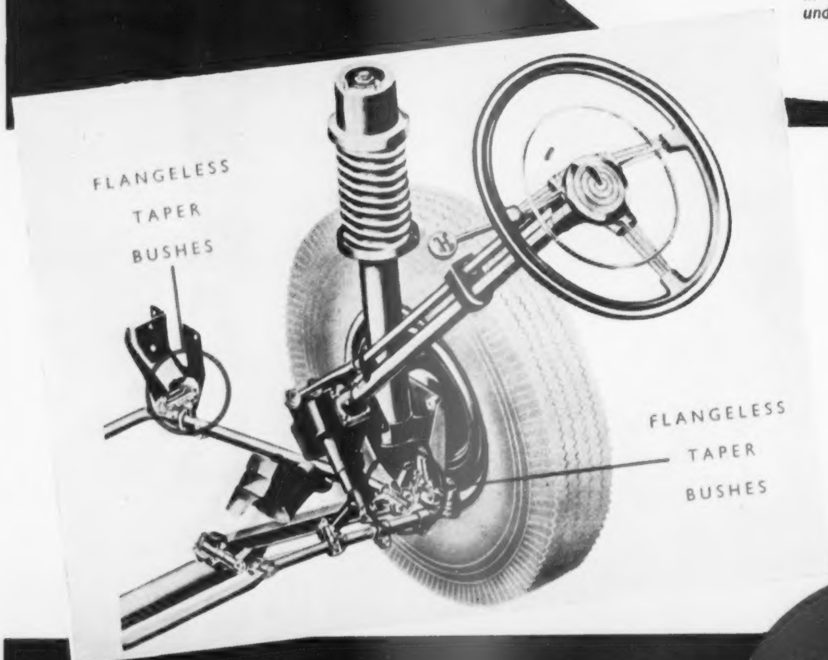
Right: Flangeless Taper Bush sectioned to show construction. In the housing flanges develop under compression.



★After exhaustive testing Ford Motor Company Ltd. now fit Silentbloc Flangeless Taper Bushes in the I.F.S. of the "Consul" and "Zephyr Six".

Left: Front Suspension of the "Consul" and "Zephyr Six".

Flangeless Taper Bushes and Frustacon mountings provide the first complete and scientifically designed rubber insulation between wheel and body.

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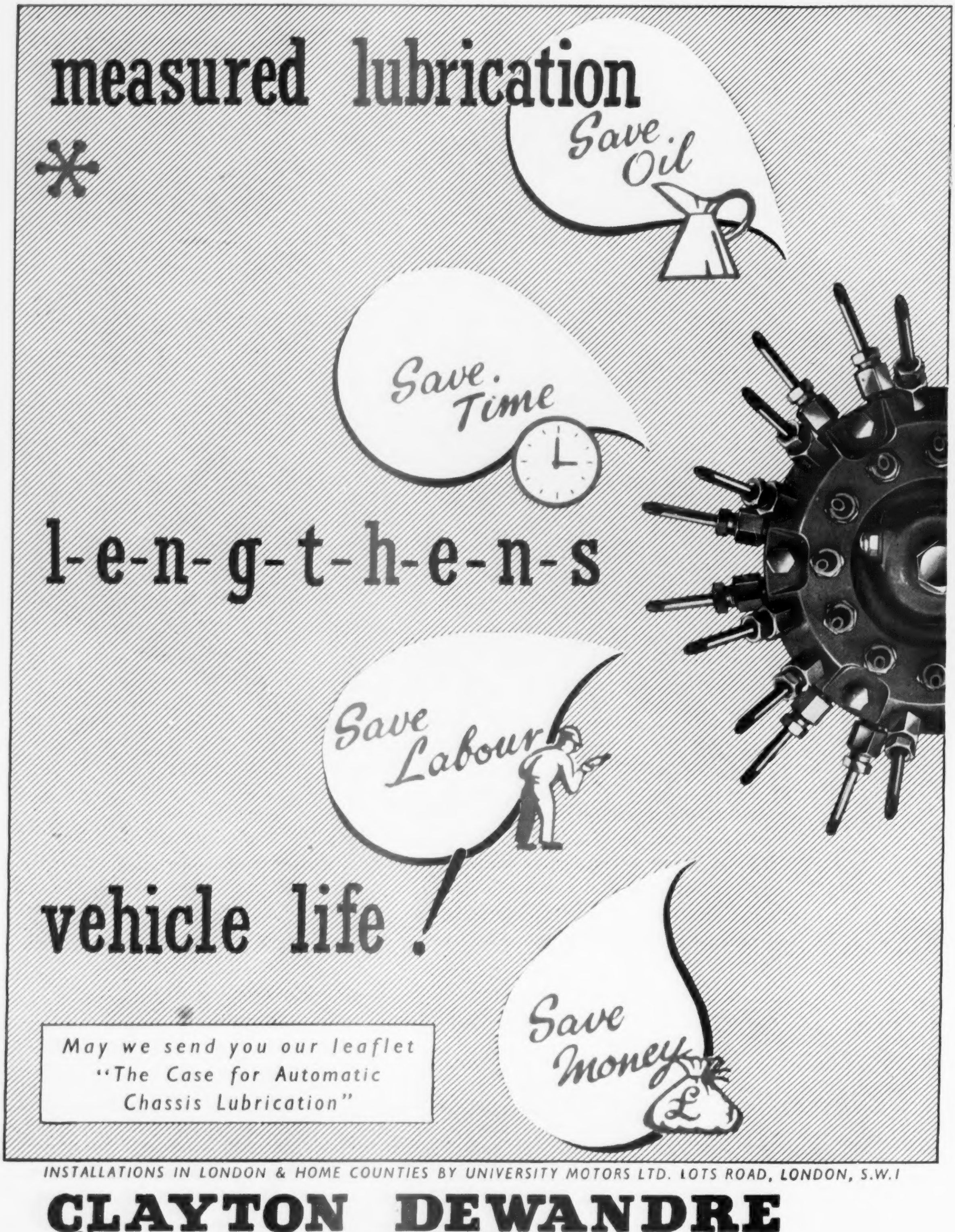
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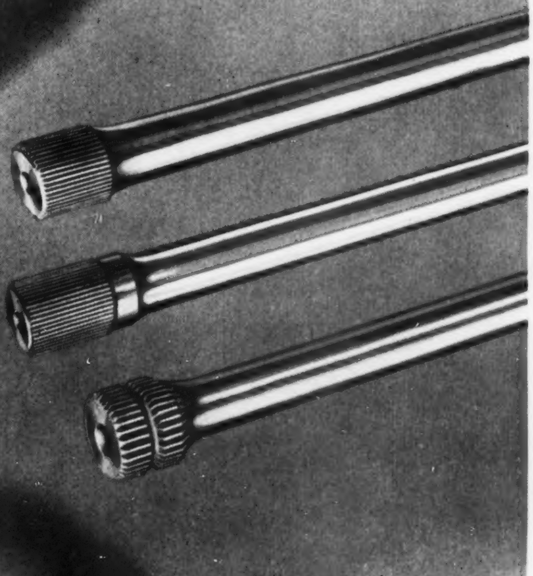

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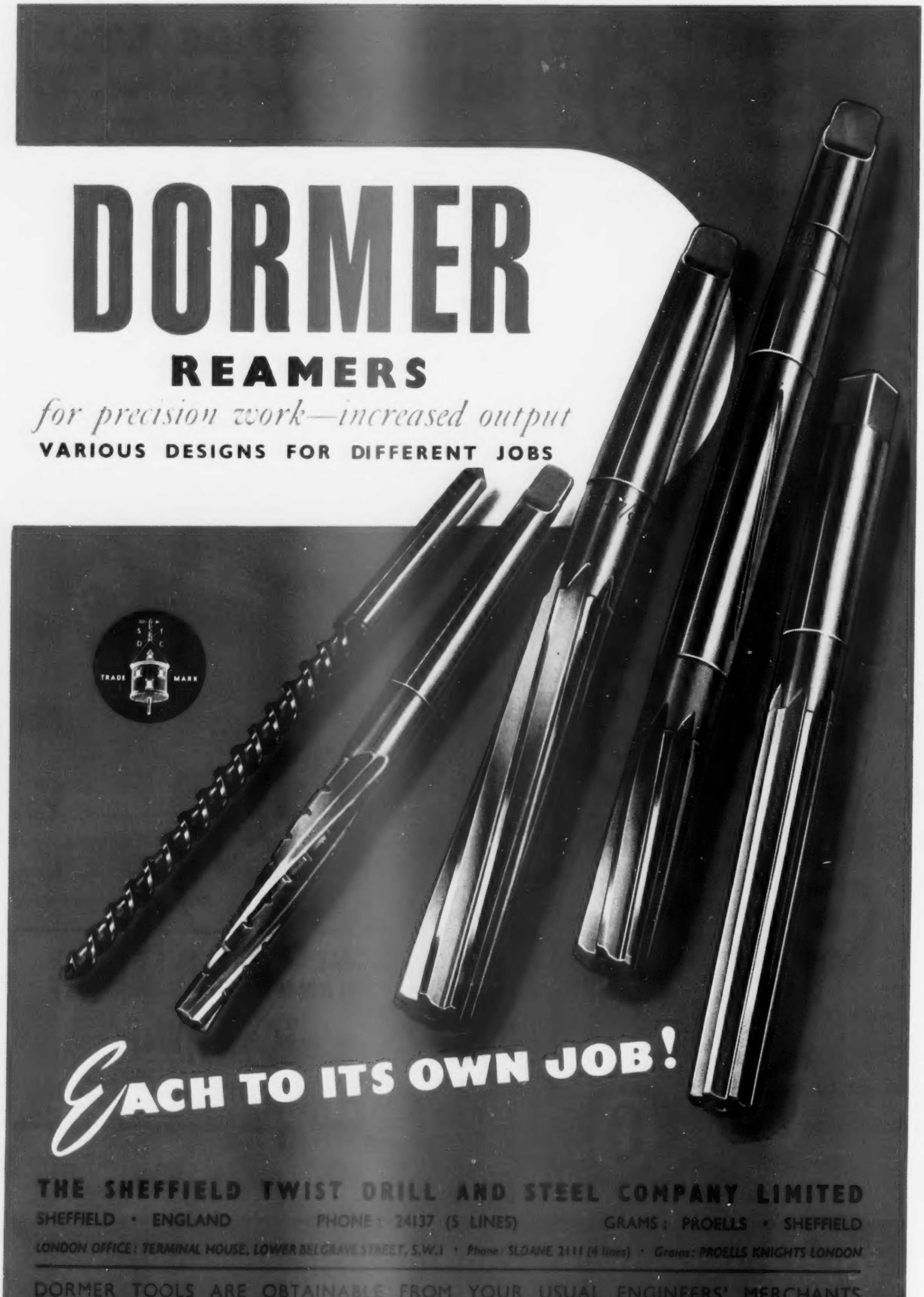
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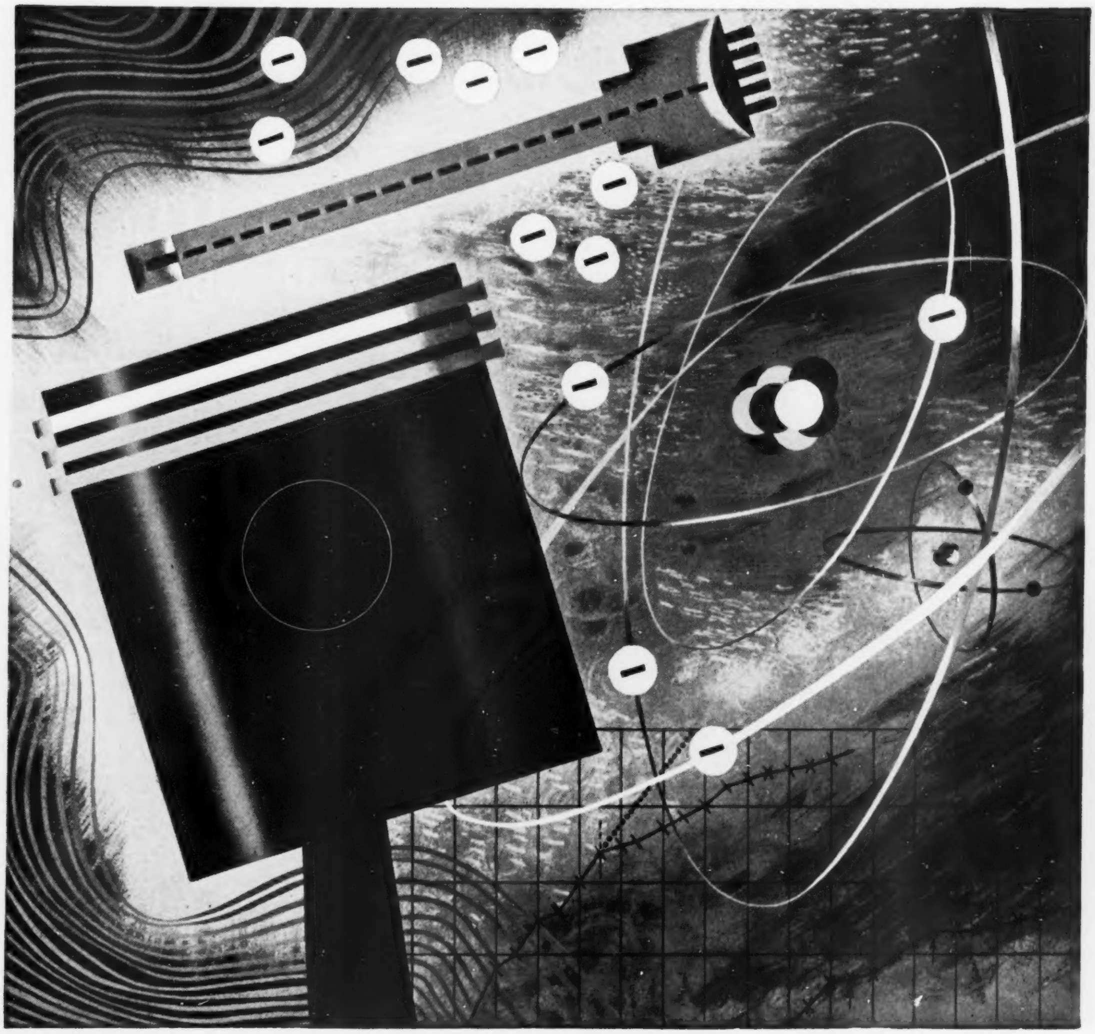

Atomic detectives

Metal wear is no new problem to engineers or research workers, but the work at Thornton has been transformed almost overnight by the application of radioactive isotopes to wear study. These are the atomic detectives of science. They record the slightest finger-print of wear with inhuman speed and accuracy.

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In another test, a length of wire is drawn vertically through a tungsten carbide die which has been made radioactive. With the aid of radioactive tracers, which are transferred from the die to the wire, the radioactivity of the wire can be assessed by means of a Geiger Counter. By this method it is possible to measure 0.00000002 grams of die wear per foot length of wire.

These are but two examples of how work on the atom has given into the hands of Shell lubrication experts research aids that will give answers in weeks instead of years. To you that means even better performance from even better Shell lubricants. To industry as a whole it means a narrowing of the time-lag between the invention of new machines and the development of lubricants to suit them.

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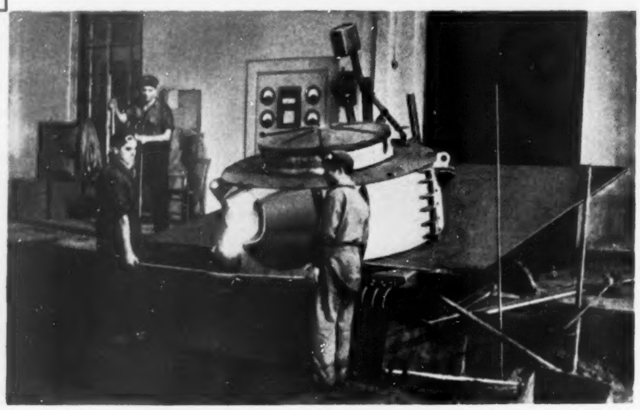
kVA	50	100	150	200	300	400	550
OUTPUT LBS. PER HR.	143	286	396	660	990	1430	1980
CONSUMPTION kWH/TON	558	558	538	508	478	467	437
TILTING METHOD	Hand	Hand	Hydr	Hydr	Hydr	Hydr	Hydr
HOW CONNECTED	1 Phase	1 Phase	1 Phase	3/2 Phase	3/2 Phase	3/2 Phase	3 Phase

The above figures which are for cold metal charges are not binding and will vary according to the product and the quality of the metal charged.



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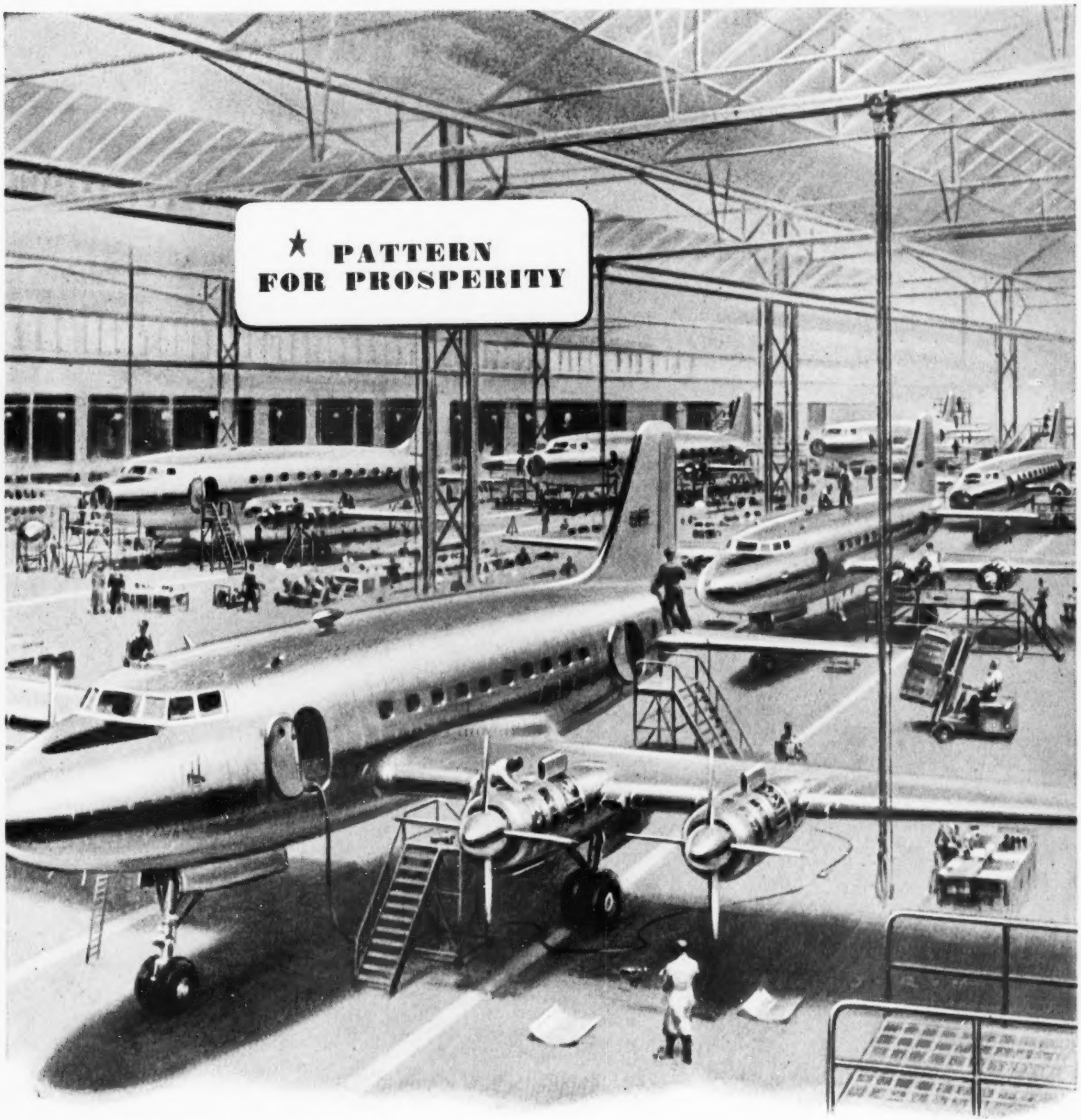

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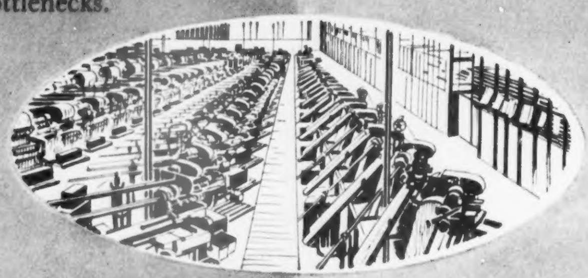

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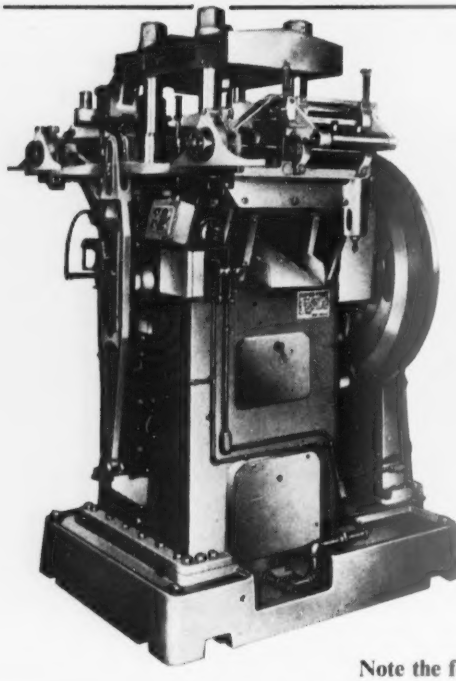
DIEING MACHINES



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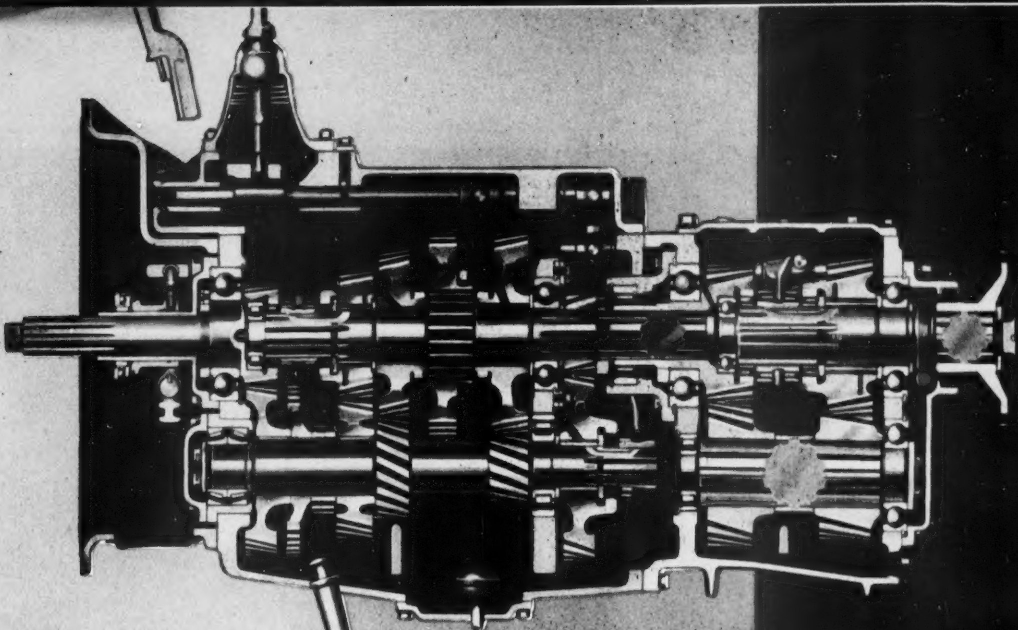
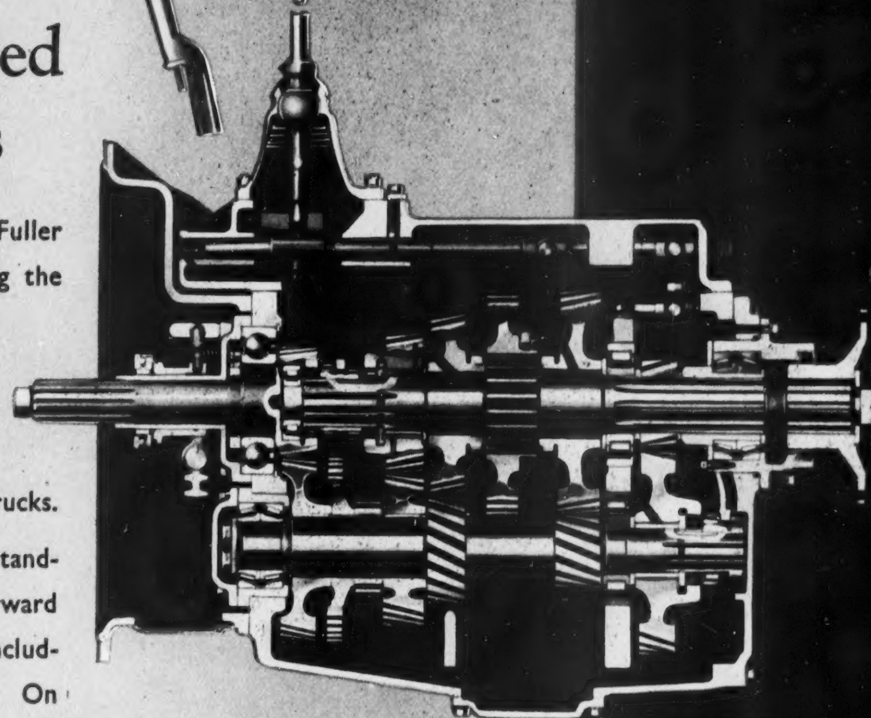


5 and 10-speed gear-boxes



The top illustration shows the Fuller 10-speed gear-box, comprising the famous five-speed box with a Fuller two-speed auxiliary box built on to it, thus providing a ten-speed box for heavy-duty operation on large trucks.

This unit is to the usual Fuller standard of high-duty, with all forward gears helical, and all changes, including reverse, by dog-clutches. On both of these boxes the gears are shot-peened and crown-shaved, to avoid stress concentration.

The lower illustration shows the Fuller 5-speed gear-box. Every gear is helical and engaged by dog-clutches and to reduce shaft deflection to a minimum the mainshaft is supported on three bearings, and the layshaft kept short, making one of the highest-duty gear-boxes ever produced.

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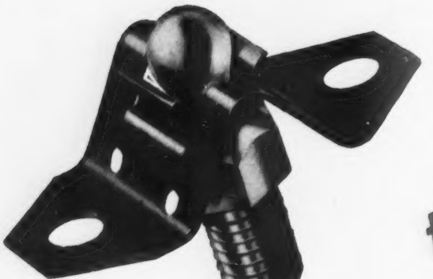

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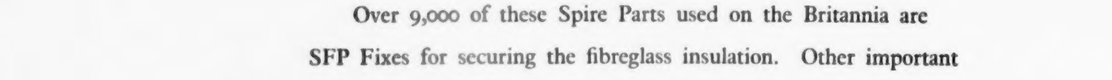

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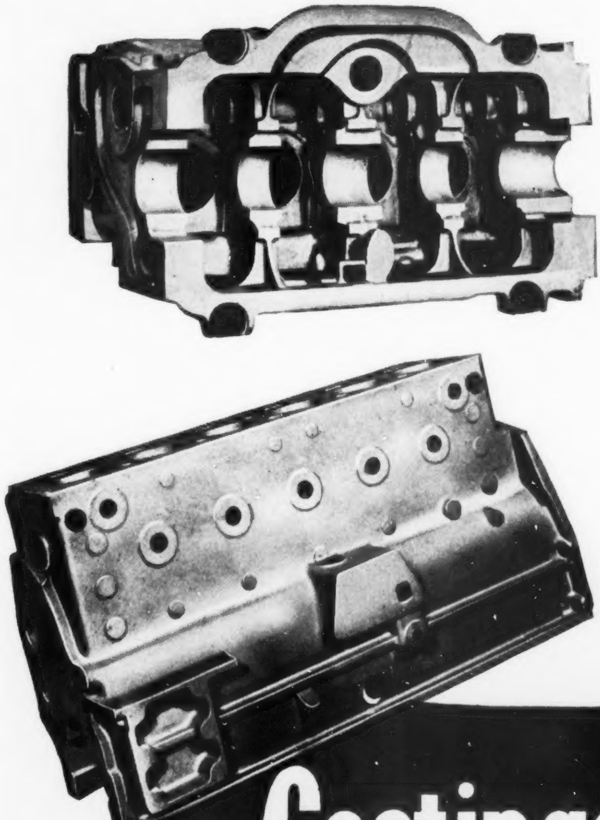

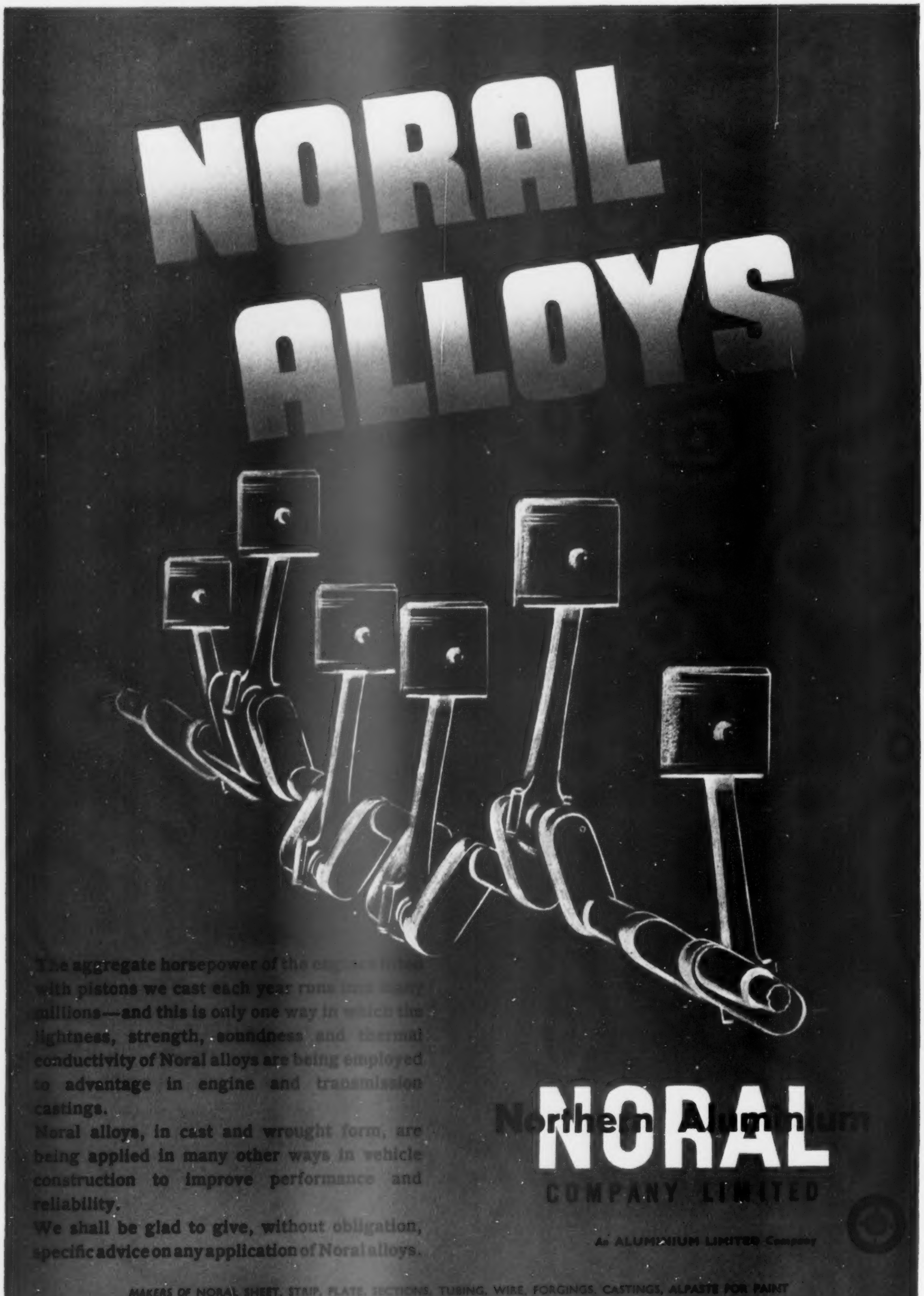
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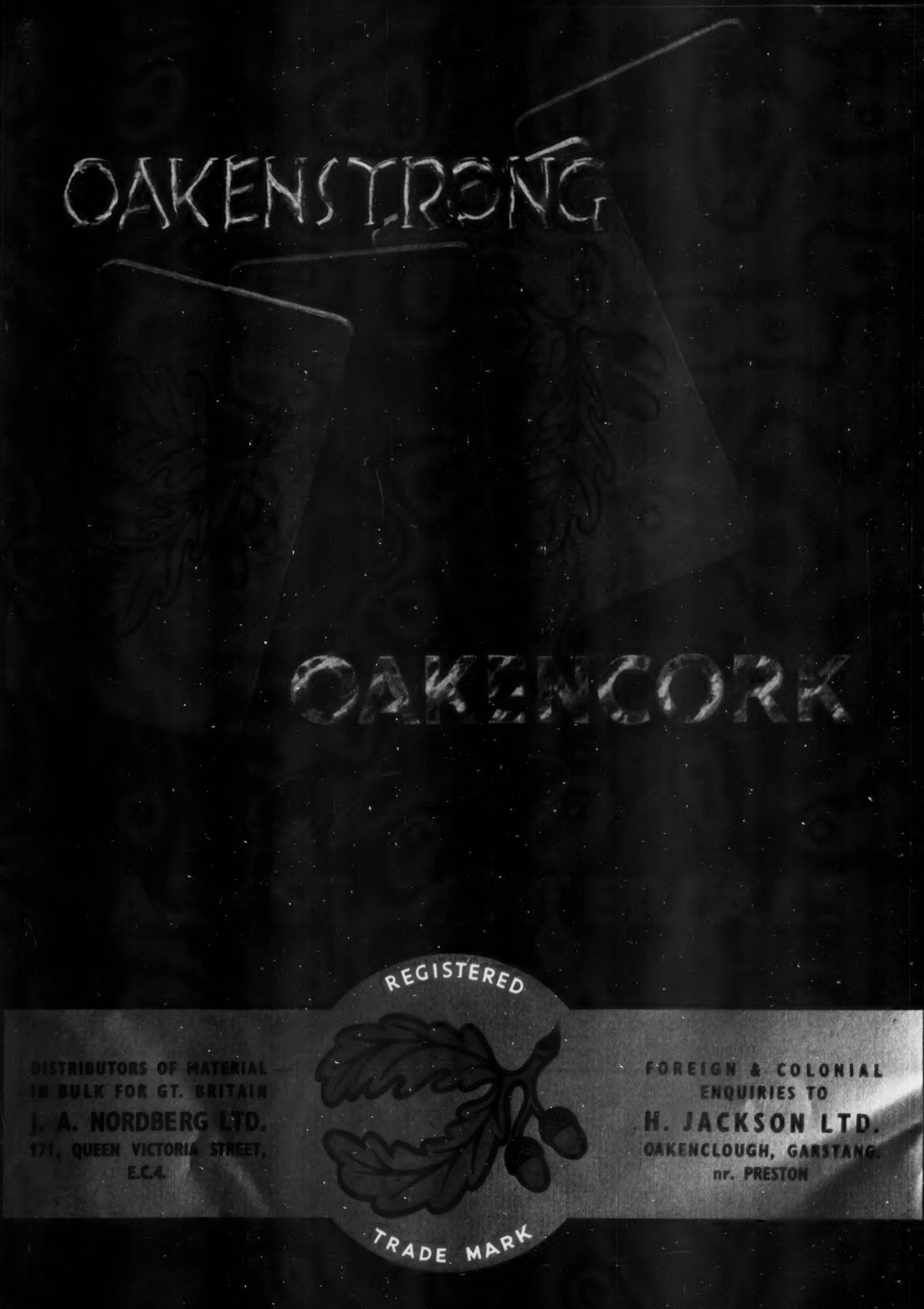
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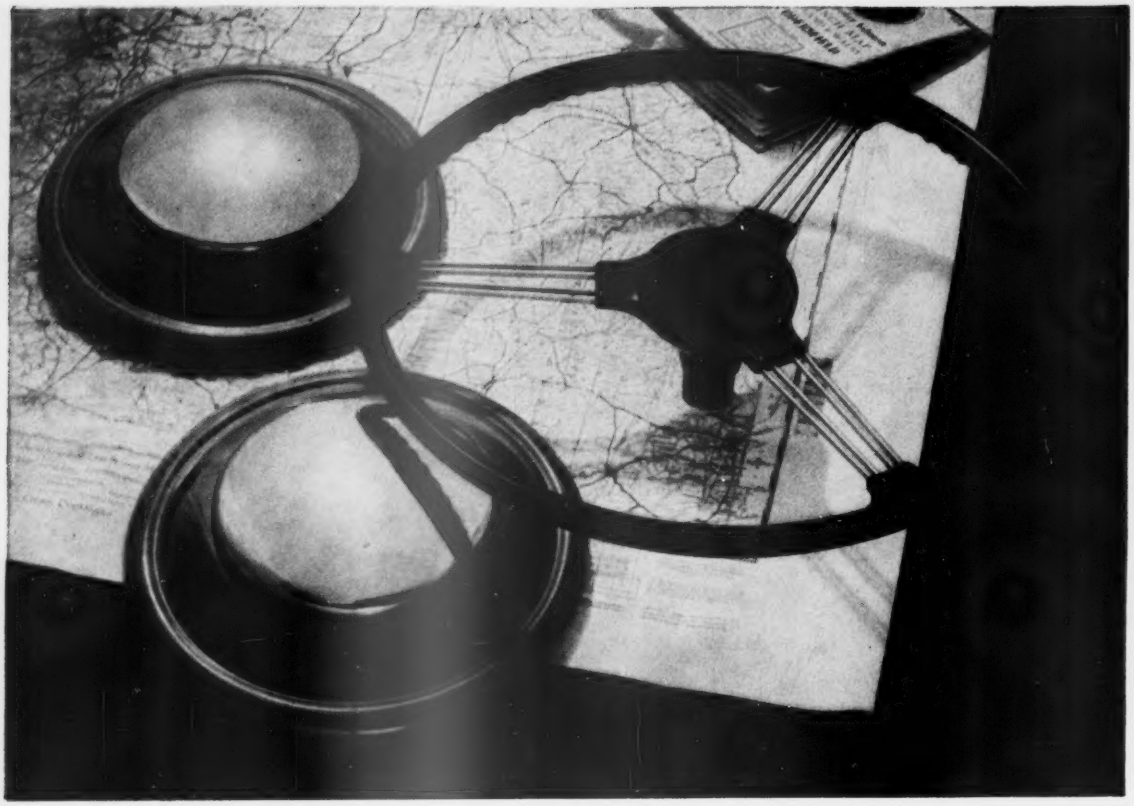

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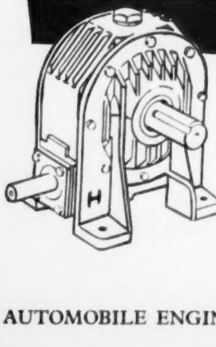

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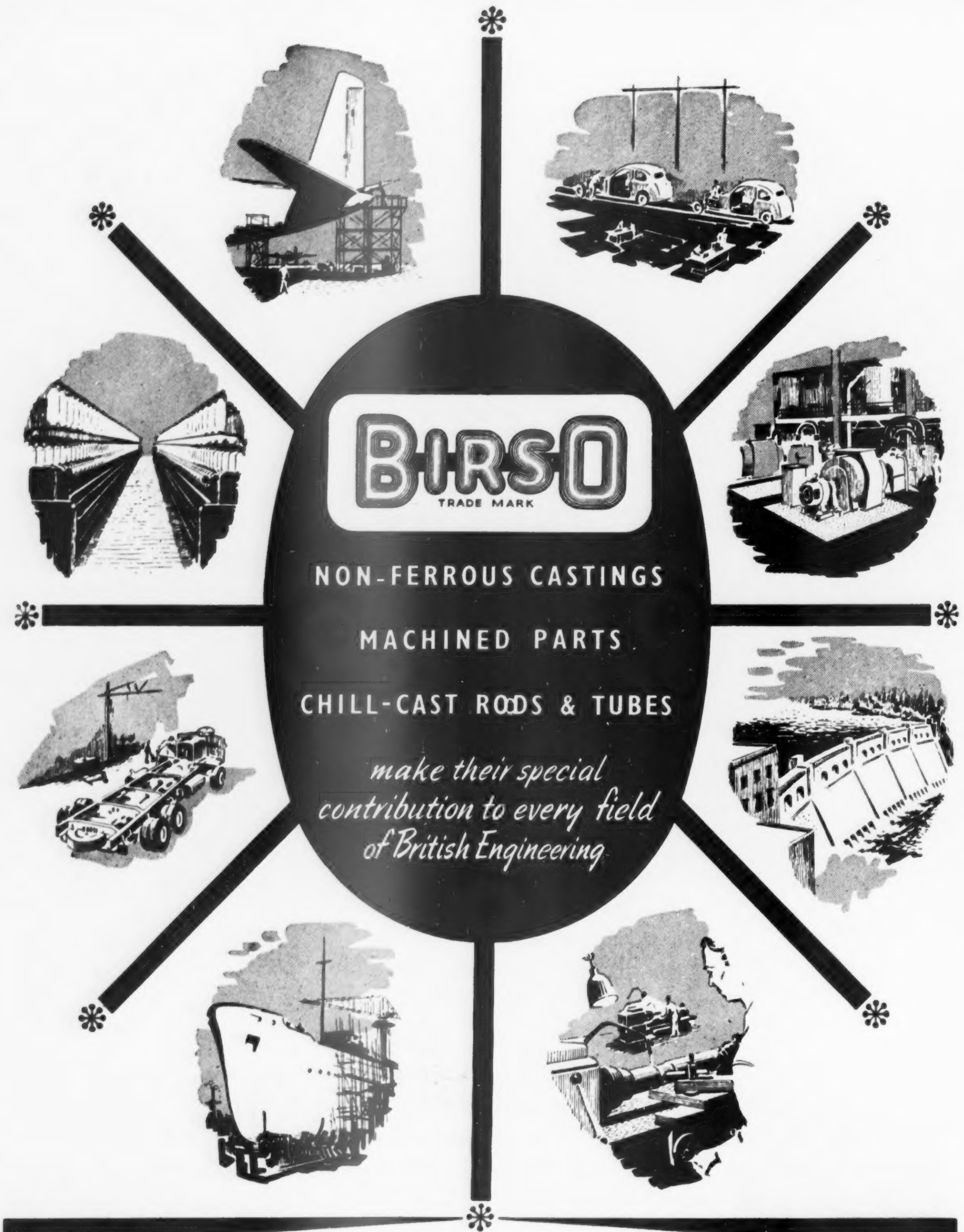
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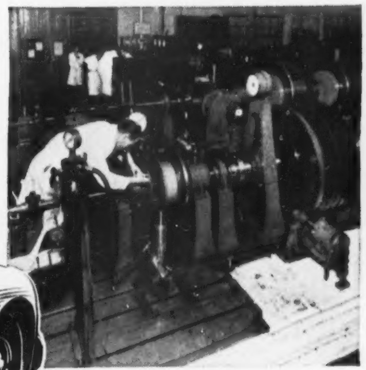

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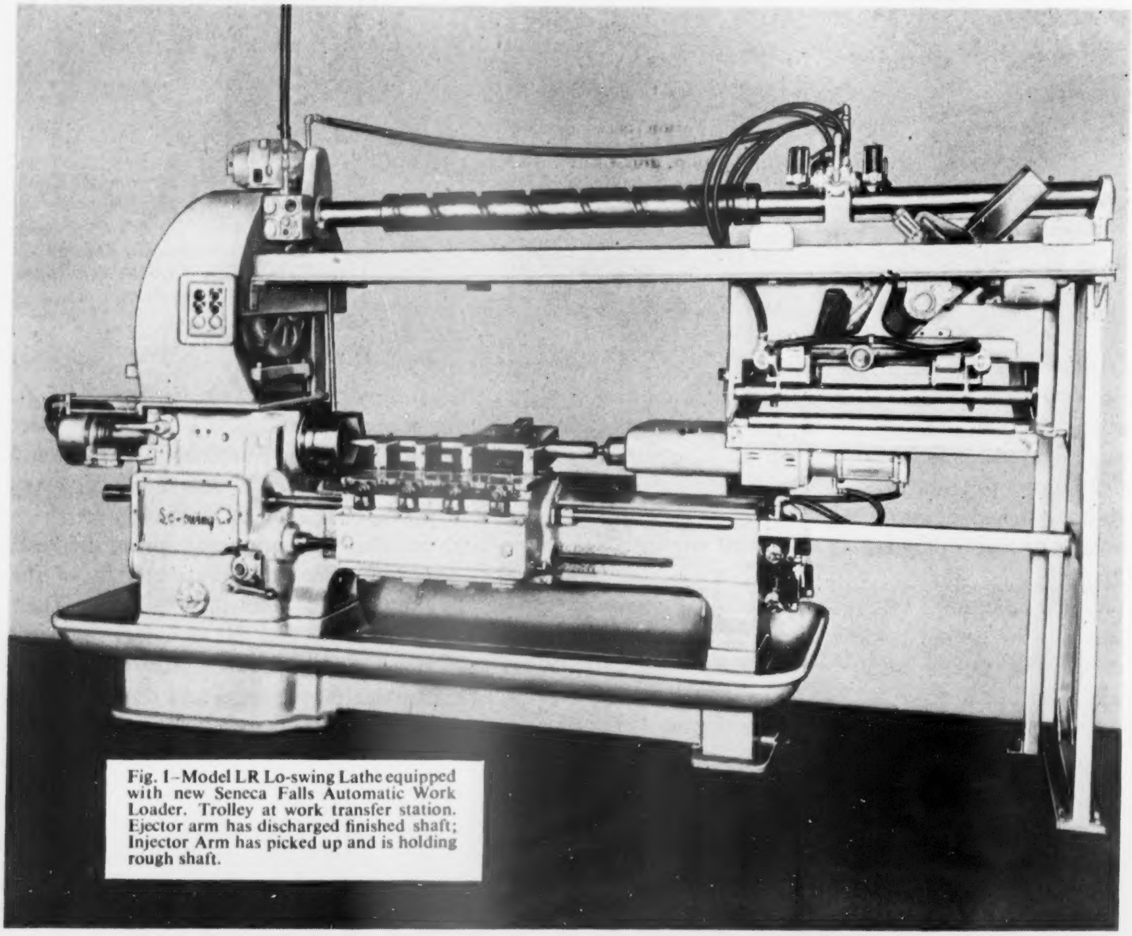

Fig. 1–Model LR Lo-swing Lathe equipped with new Seneca Falls Automatic Work Loader. Trolley at work transfer station. Ejector arm has discharged finished shaft; Injector Arm has picked up and is holding rough shaft.

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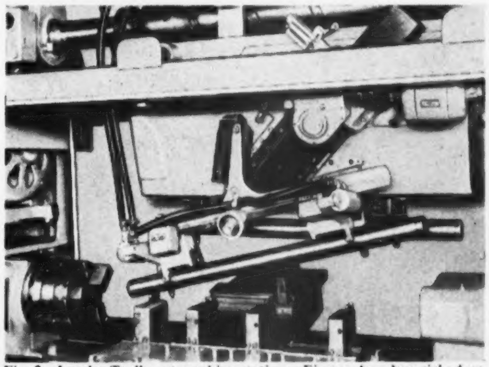

Fig. 2—Loader Trolley at machine station. Ejector Arm has picked up finished shaft. Injector Arm is tipping and lowering a rough shaft. Headstock spindle is stopped, chuck jaws open and tailstock centre retracted.

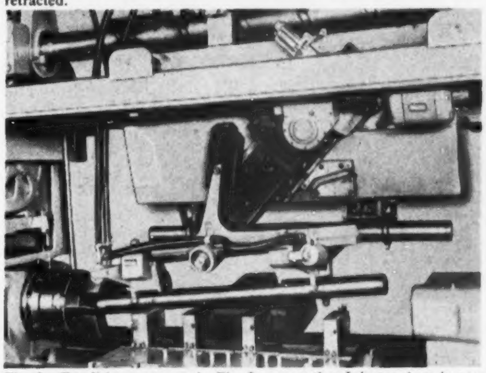

Fig. 3—Conditions same as in Fig. 2, except that Injector Arm is now bringing rough shaft to the horizontal position for entry into chuck jaws. Ejector Arm still holds finished shaft.

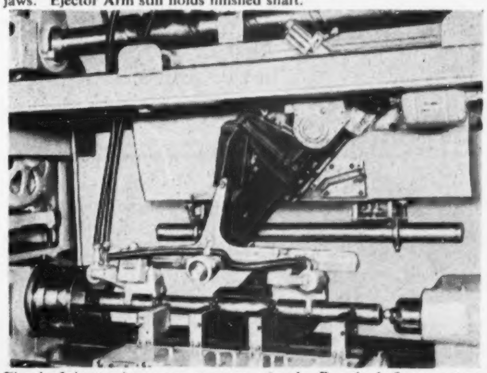

Fig. 4—Injector Arm movement completed. Rough shaft on centres, chuck jaws closed and tailstock spindle advanced. Injector Arm Fingers will next release shaft, and machining begins while Trolley discharges finished piece and returns with the next rough shaft.

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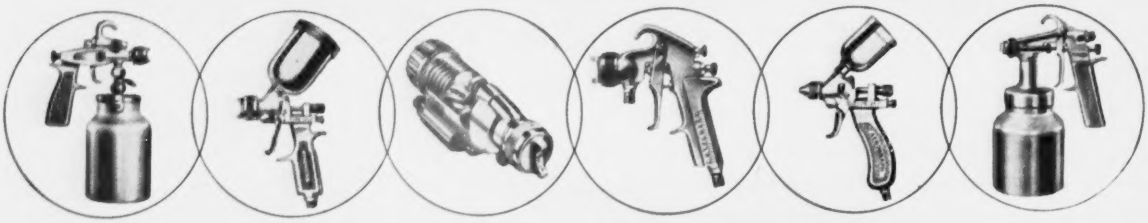

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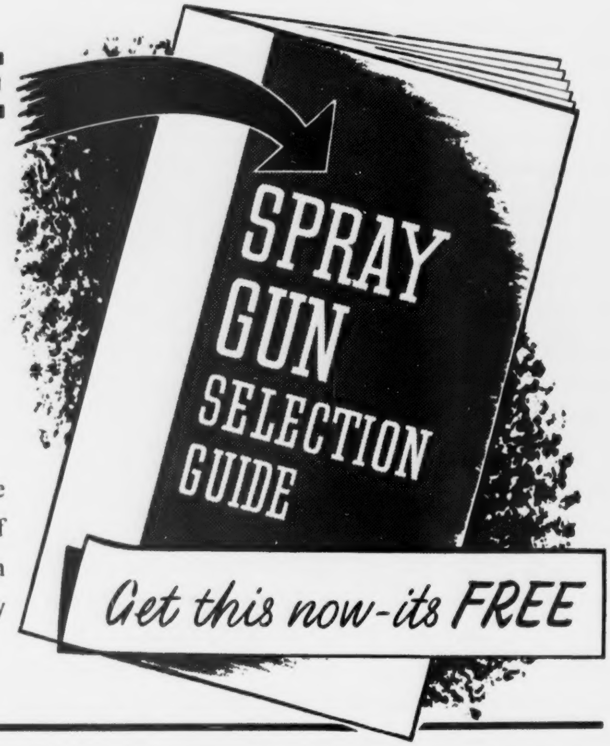

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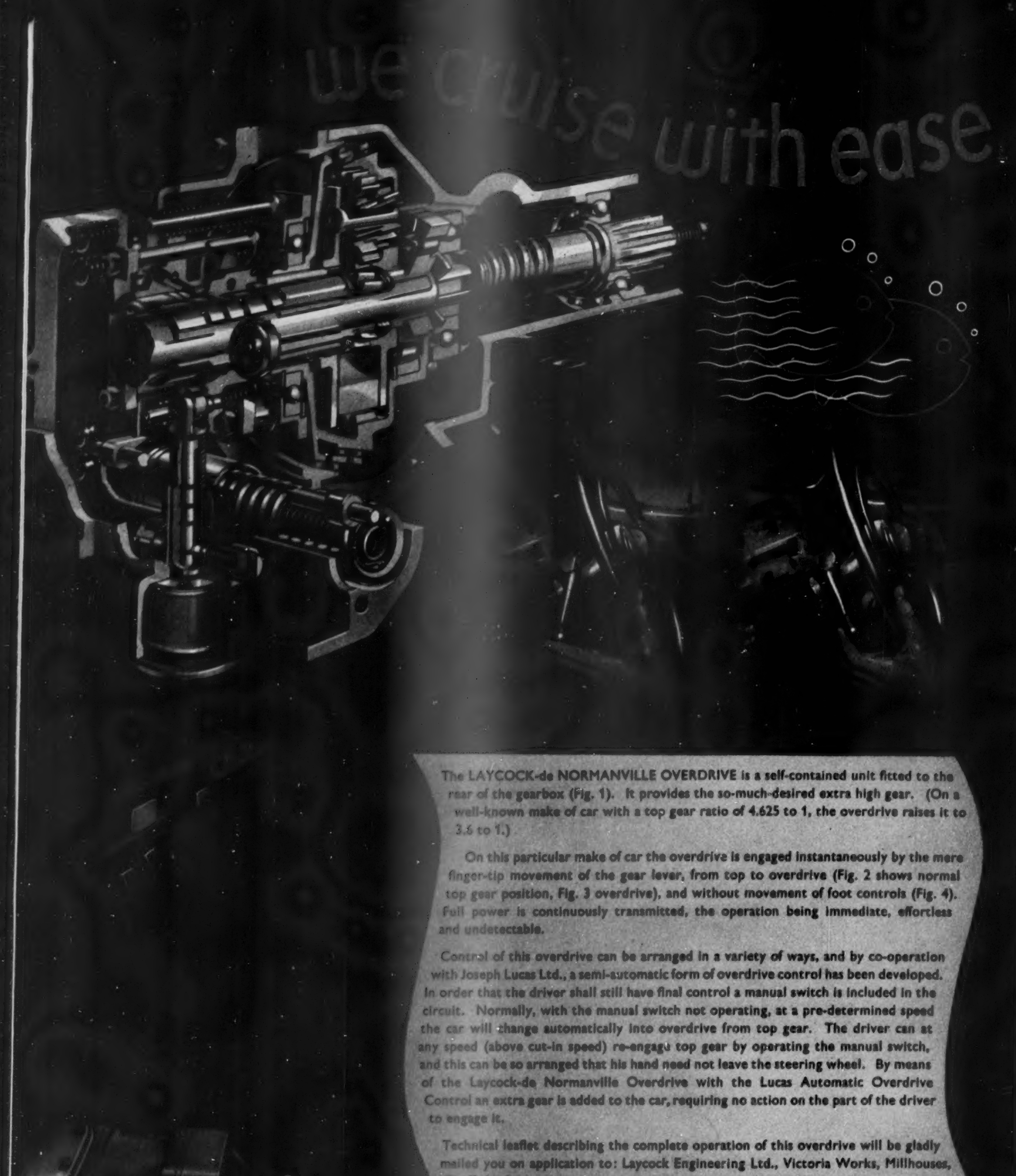
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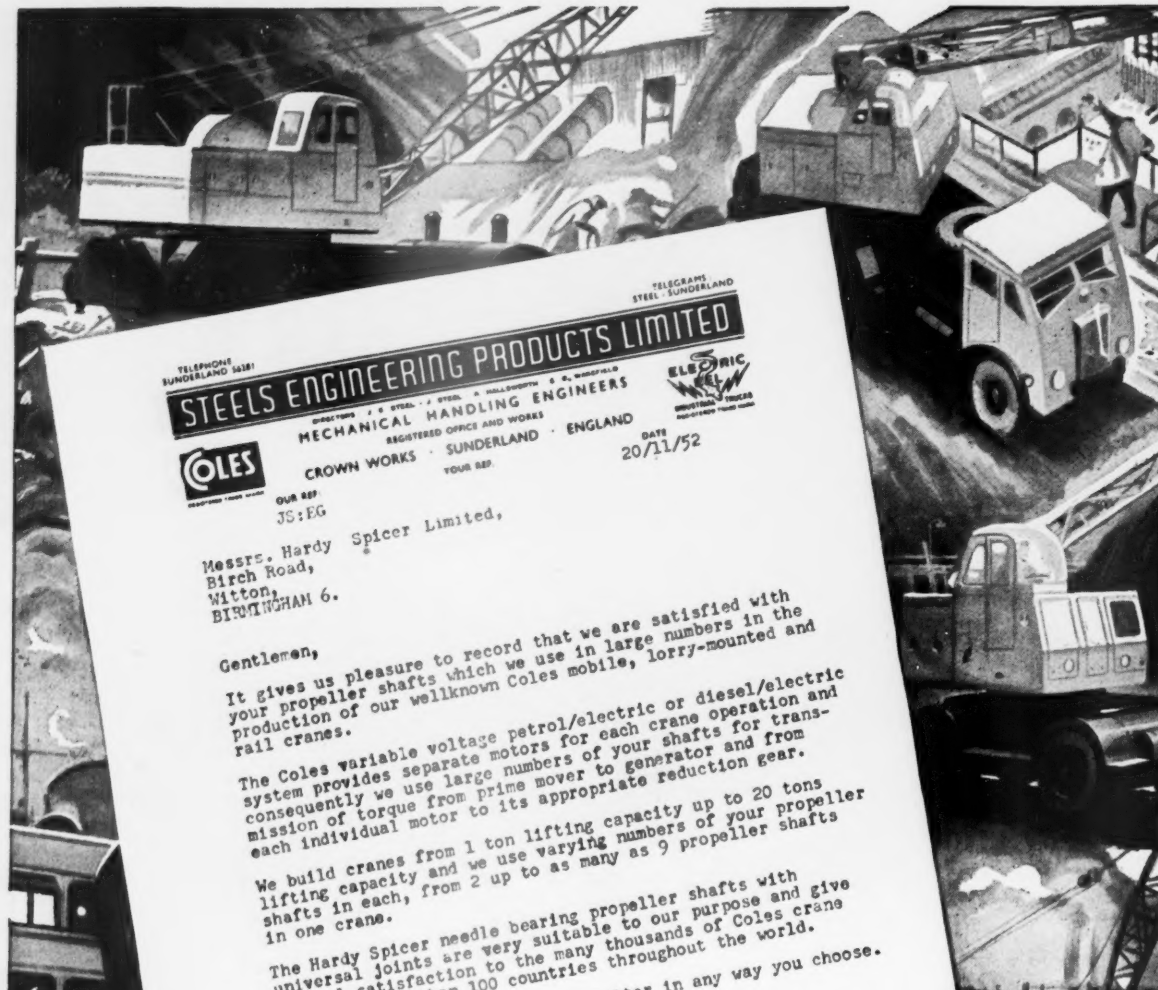

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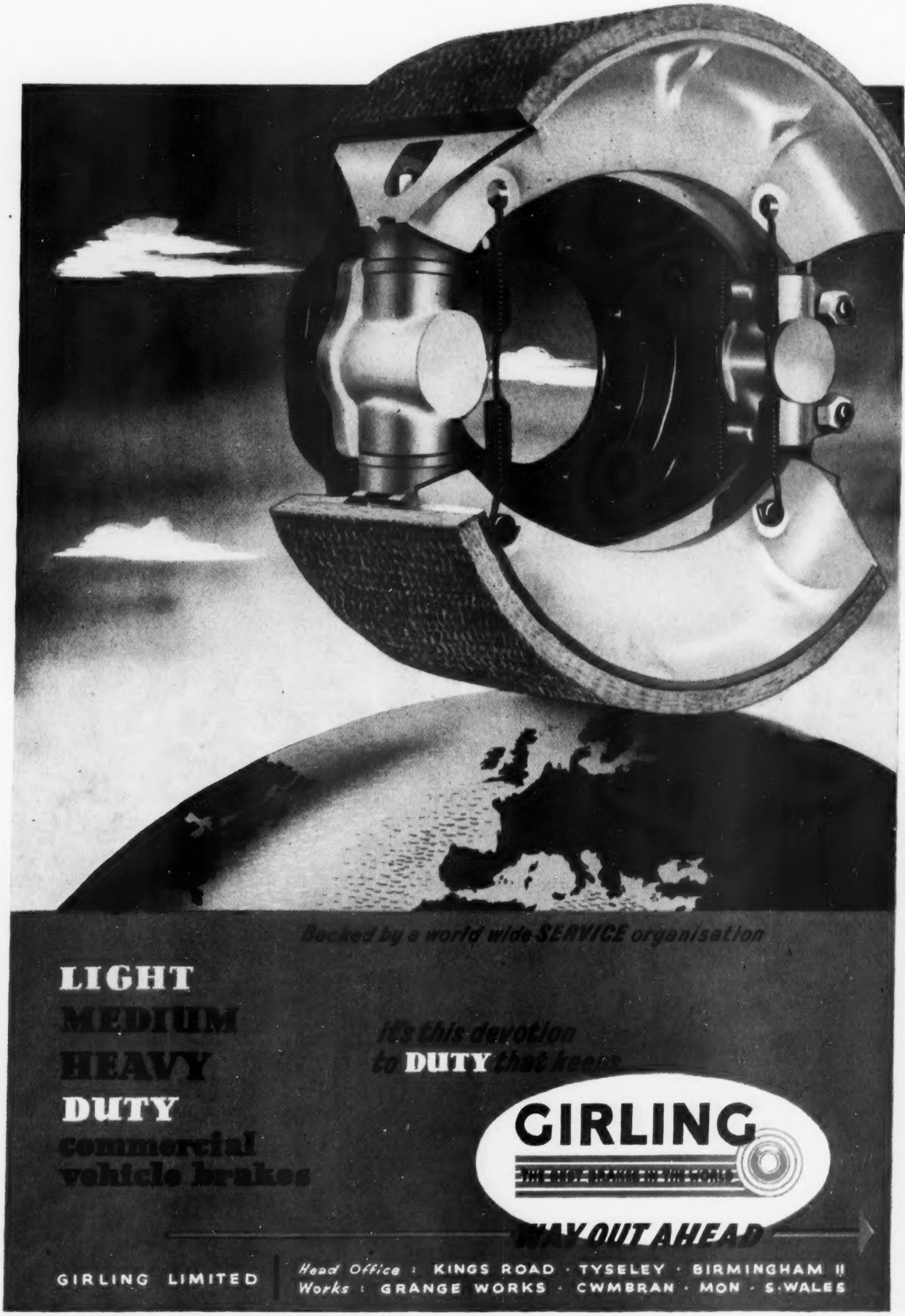
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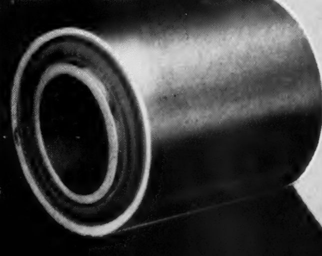

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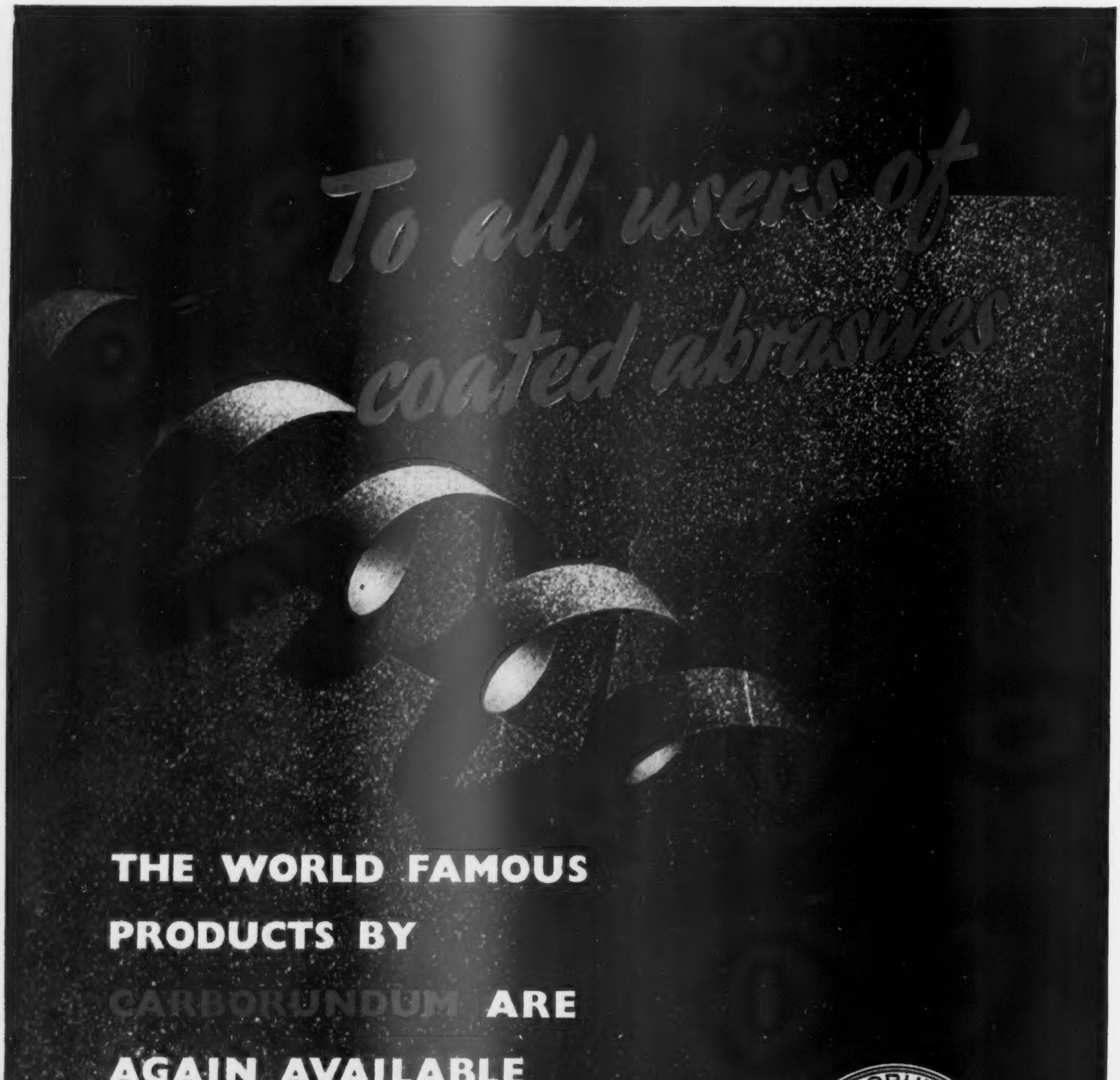
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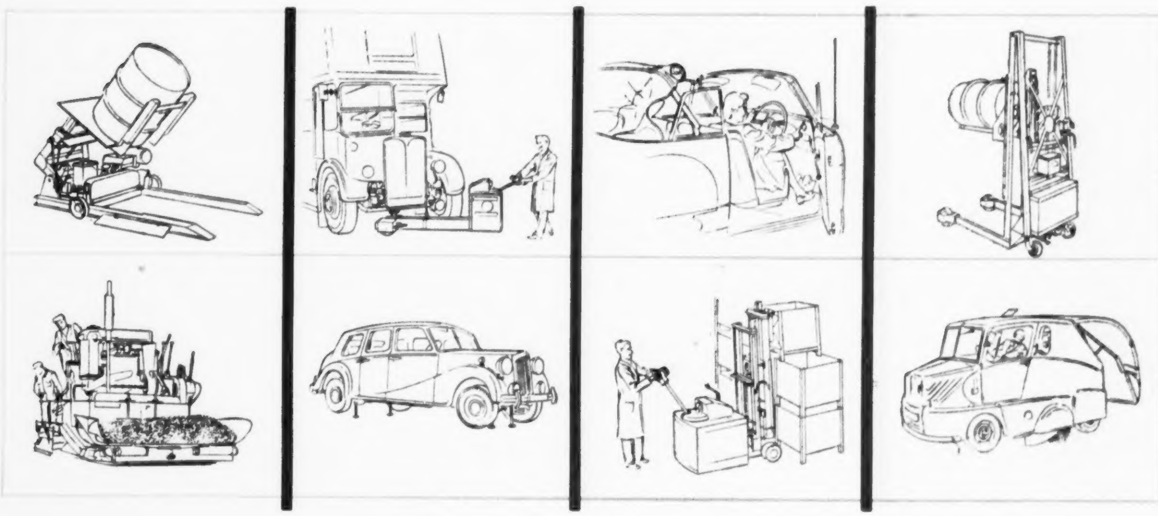

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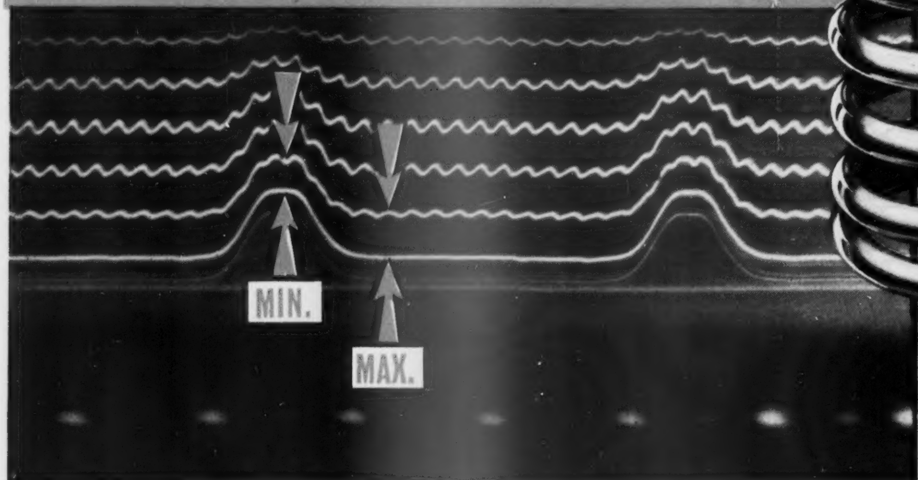
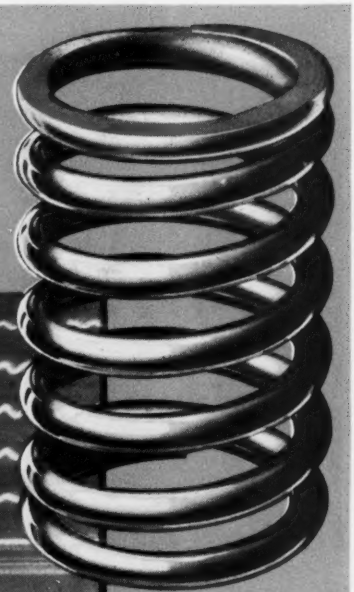
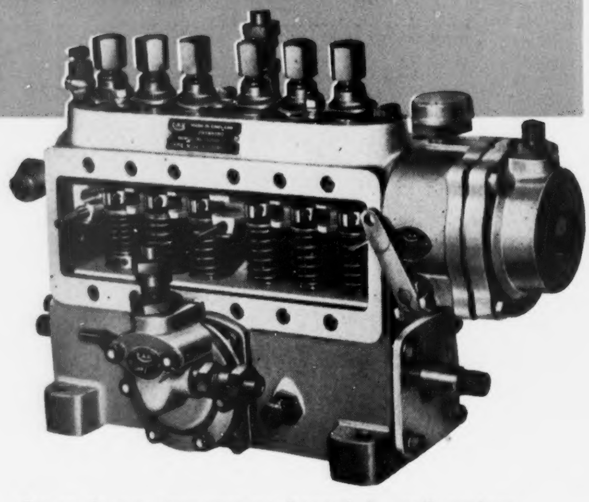




The firm line shows the cam rise over two cycles. The lower dotted trace shows time intervals of 10 milliseconds.



When a spring is abruptly compressed, its coils may be caused to "surge", that is, the centre coils are displaced relative to the ends to a greater extent than during the slow compression of the spring, and then vibrate.

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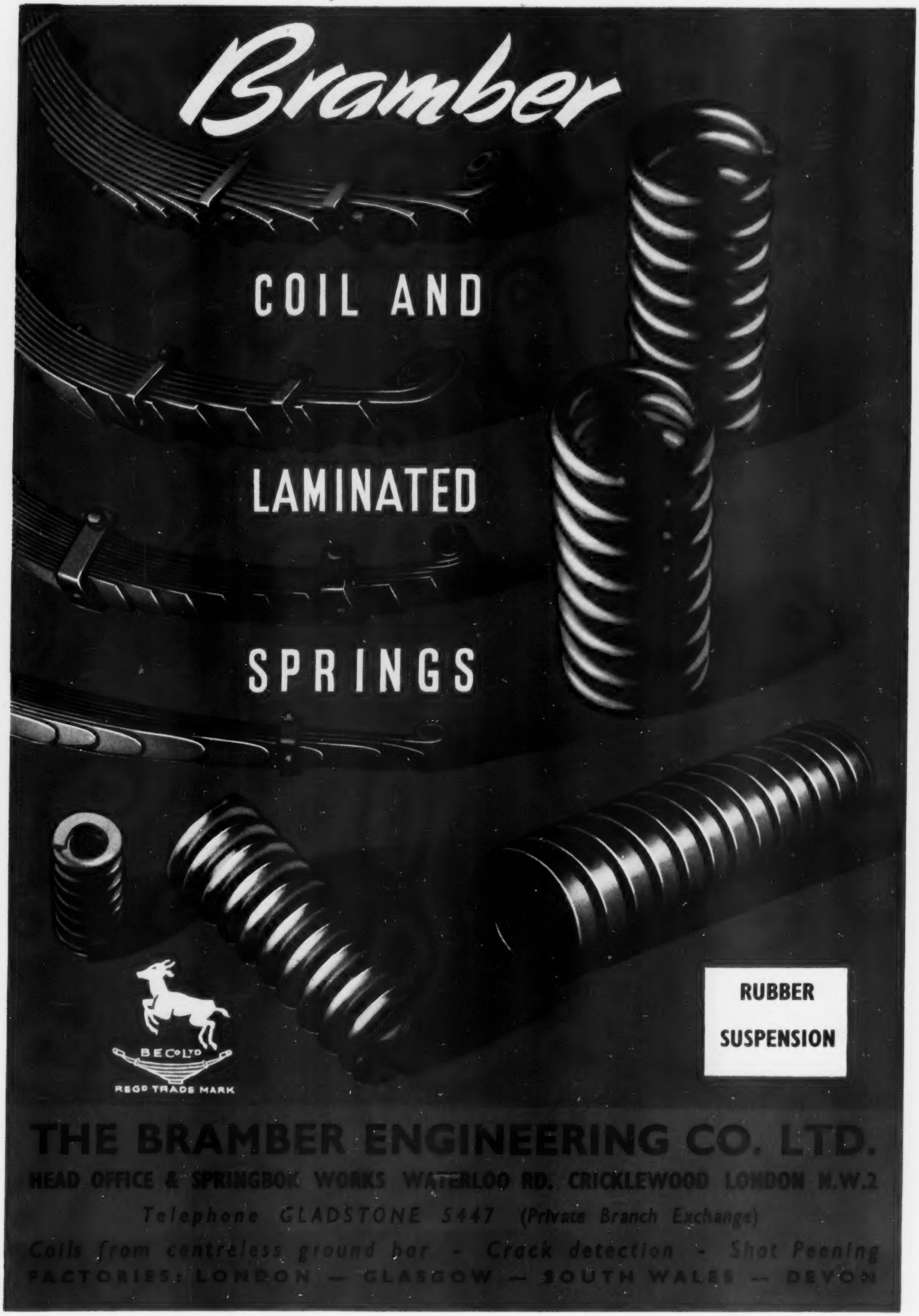
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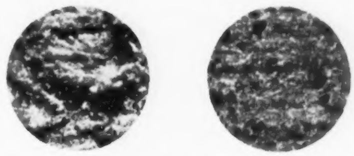

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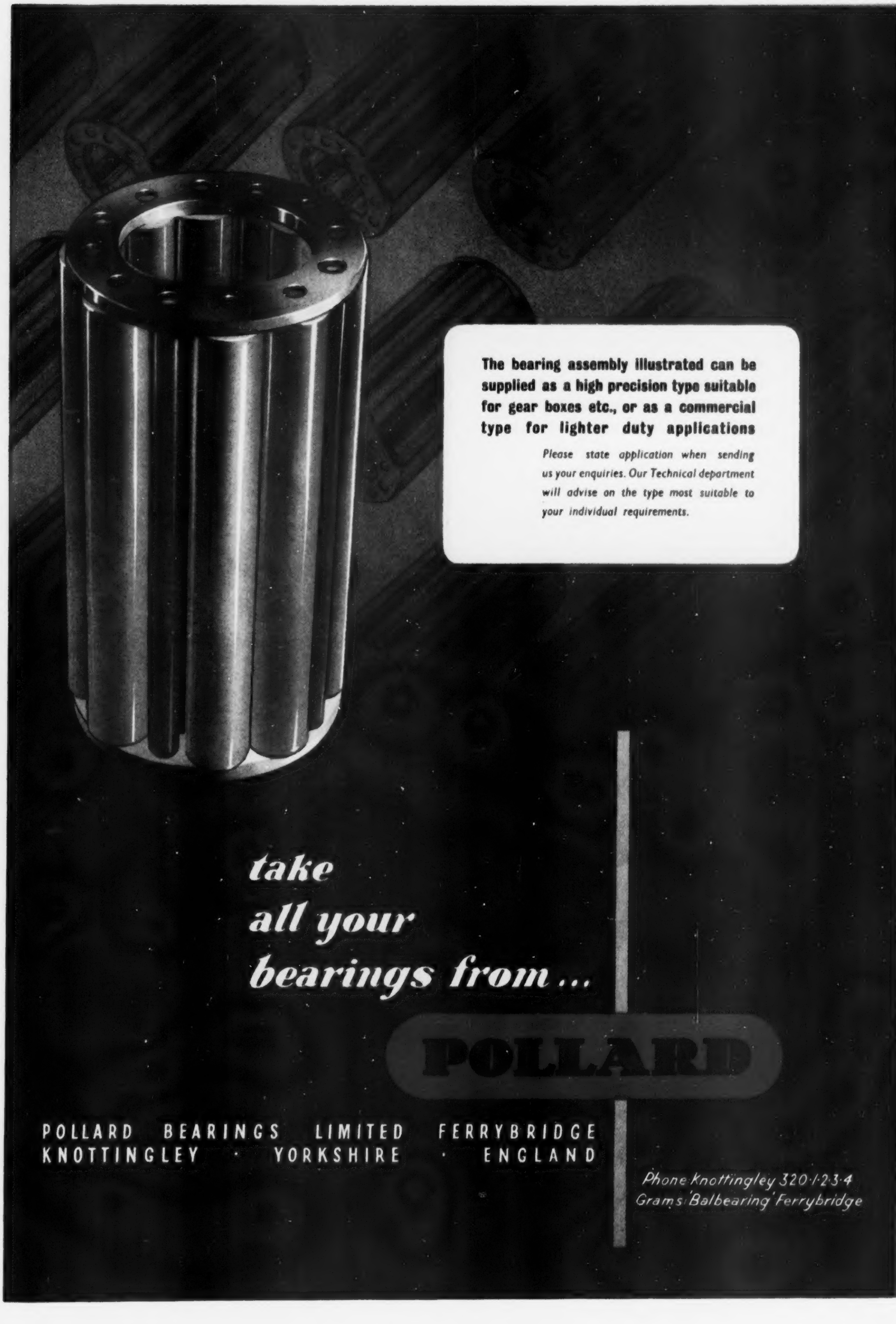
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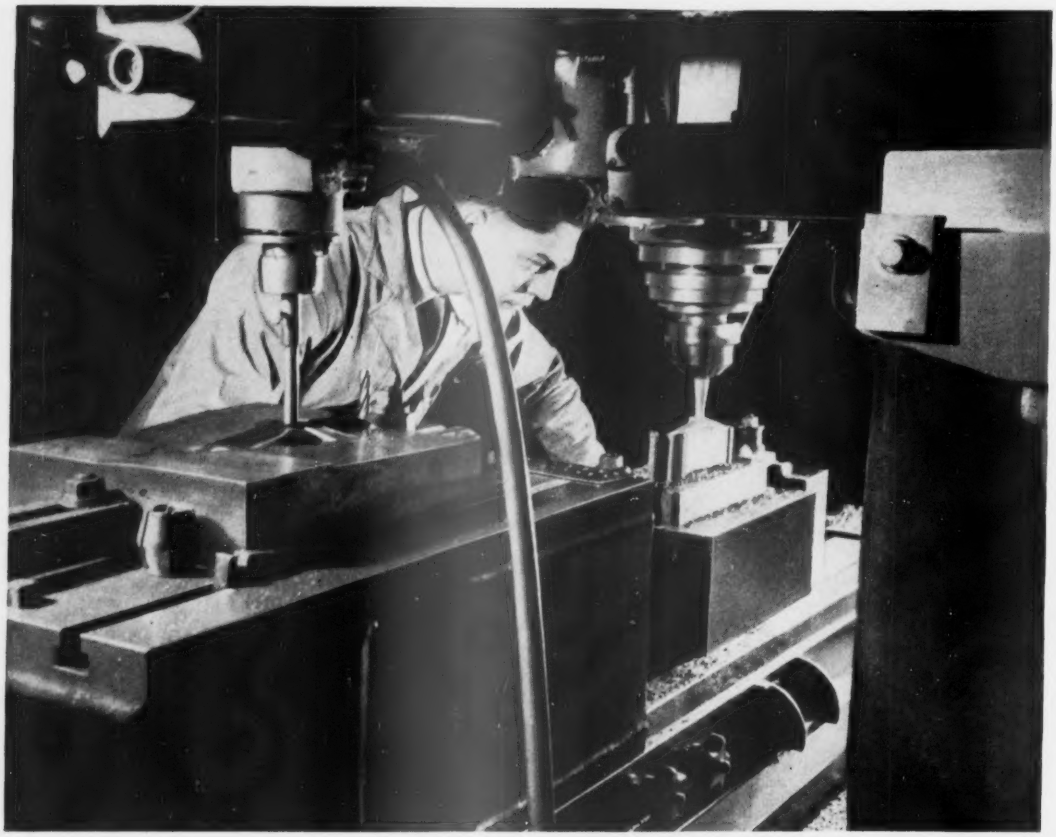

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AUTOMOBILE ENGINEER

Design, Materials, Production Methods, and Works Equipment

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Nomenclature

WHILE manufacturing developments in the automobile industry are all towards greater precision, there is still a disturbing and perhaps increasing imprecision in nomenclature. This may seem to be a relatively trivial matter, but it does lead to considerable confusion and often to waste of effort as well. In some cases one part will be known by completely different and unrelated names; in others a single designation may have different connotations as between one organization and another.

Incidentally, the above remarks apply to industry in the United States of America as well as to British industry. For example, the President of the Conveyor Equipment Manufacturers Association recently stated that the confusion of names for machines and parts cost American industry millions of dollars a year and led to Tower of Babel confusion. In defence contracts it was frequently found that as many as five different terms were used for the same piece of equipment, while on the other hand one designation could mean five different things to five organizations. To allow the correct designations to be used for conveyor equipment, a dictionary has been prepared and gives more than 1,500 terms and definitions agreed upon as standard terminology by the members of the conveyor association.

Coachwork

In this country, coachwork nomenclature is an outstanding example of lack of standardization. It is a good many years since any recommendations on this subject were made by an acknowledged authority, and considerable confusion still exists. There is urgent need for a thorough review of body types and their classification in a comprehensive list giving titles bearing a distinct relation to their functions. Moreover, as each year brings further additions to the range, the problem will become worse unless it is soon dealt with.

It is not out of place to remark here upon the wide diversity of body styles that any recommendations must embrace, and the comments now made can be merely pointers. Particular comment may be made upon the type of body that, more than any other, carries a title that causes much confusion. We refer, of course, to the coupé. To the average motorist this name indicates a body with a folding head, but in actual fact it can refer to either a folding head or a fixed head type. To the coachbuilder, the designation coupé is not sufficiently definite. The recommendations issued over twenty years ago classified these bodies as the coupé cabriolet and the coupé respectively. Something better than this is needed.

Although the word cabriolet may bring pleasant memories to the older generation of motorists and coachbuilders, it no longer conveys a true indication of the style of body to which it refers. It is in fact rarely used, and from the variety of words now used to denote that the coupé in question has a folding head, the words "drop head" or "convertible" would appear to be the most appropriate. Either term will do, but a definite choice should be made.

Few bodies are now built in purely two/three seater style, the type to which, when glass side windows were fitted, the word coupé was originally applied. It would therefore seem desirable to include some reference to seating capacity, although this may be a matter for the manufacturer rather than for the recommending authority.

Sports cars

Whether the designation "sports" should be used in association with the description of a car depends mainly upon the type of chassis concerned, but some consideration should be given to the type of body that is involved. In the past there have been cars described as "sports limousines" which is, to say the least, an incongruous title for a type of body normally associated with stately social occasions.

Nomenclature for component parts of bodies also needs clarification, for with the increasing use of all-metal coachwork, body parts have assumed new functions. This is particularly the case with pressed metal bodies where the close affinity with American practice has led to the introduction of some unusual terms. To-day, a coachbuilder making use of standard metal body parts is often confronted with a parts list that is far from self-explanatory. Either the items have to be examined, or the drawings studied, before the true functions of the parts are apparent. It is not suggested that American nomenclature should be adopted in its entirety, but many American terms are much more descriptive than British terms, and since many of them are already fairly widely used, it would be wise to make their use general.

That time and money are wasted through the confusions arising from the failure to apply a standard nomenclature is beyond dispute. The revision of present practice and the preparation of a standard list is a matter that should engage the early attention of the several associations that

possess the knowledge and authority to formulate and advise on a comprehensive schedule. This is not a matter that will arouse the conflict of interests that occurs when attempts are made to standardize dimensions, and therefore it should not be difficult to reach agreement.

Leaf Spring Design

TO simplify, make understandable, and yet provide an accurate basis for the study of a complex subject is not an easy matter. Those of an older generation who were taught from "*French Without Tears*" and "*German Without Tears*" know this only too well. Despite their titles, those excellent books promoted tears rather than proficiency in the designated languages. That it can be done was proved by Sylvanus P. Thompson with his "*Calculus Made Easy*". Countless engineers owe what proficiency they have in the calculus to that estimable little classic. Many found that mastery of its contents gave them sufficient knowledge of the subject for their needs; others found that it supplied an invaluable introduction to more advanced standard works.

The above observations arise from a persual of a booklet, "*Leaf Spring Design*" by Alan Hodgson of Richard Berry and Son. It deals simply, comprehensively and very clearly with a subject of fundamental importance to all chassis designers, since the springs have a considerable effect both on riding comfort and on the insulation of the chassis structure against sudden road shocks.

There are few chassis design engineers who would not profit from a careful study of this work, which deals comprehensively and yet with great economy of words with the fundamentals of leaf spring design. Any designer who masters the contents will be able to comfort himself that his knowledge of an important component has been widened, and application of the knowledge will materially assist in providing optimum service conditions with optimum economy.

A limited number of copies is available gratis to senior executives in the automobile industry; application should be made to J. Brockhouse and Co. Ltd., 25, Hanover Square, London, W.1. The value of this work is such that it should also be in the hands of the juniors of to-day who will become the senior executives of the future. It is to be hoped that the Brockhouse organization will consider the possibility of producing further editions, perhaps in a less expensive format and not necessarily for free distribution.

Production Figures

ABOUT a year ago the automobile industry came under fire from several quarters because exports were falling off. All informed persons knew that this was due to governmental restrictions of imports and not because British cars were no longer competitively priced. Nevertheless, there were suggestions that steel, then in short supply, should be diverted from the automobile industry to other purposes. This would, of course, have been a policy of despair since it would have reduced plant utilization with a consequent detrimental effect on costs.

Fortunately, wiser counsels prevailed, and although material shortages continued to hamper production, factories continued to run at a fairly high percentage of full productive capacity. More recently, the material supply position improved and the members of the automobile industry have reason to be proud of the results they have achieved this year.

Passenger car production in May 1953 was the largest ever attained in this country. The total output was nearly 50,000 units, an increase of 4,000 units over the previous peak month, November 1950. There is a similar tale to tell of commercial vehicle manufacture. Almost 19,700 commercial vehicles were produced in May, an increase of 300 per week on the previous average for this year, and nearly 3,000 more than the April total.

Not only has total output improved, but despite the end of the buyers' market, there is now a very encouraging trend in export trade. For example, in May nearly 31,000 cars went to overseas markets. This is almost 8,000 more than the monthly average for the period January-April of this year. Truck and bus exports also improved appreciably in May.

Perhaps the most encouraging sign is that Canada was the principal overseas market during the month of May, when that country took 5,685 passenger cars valued at £2 million. In addition, 3,064 cars went to the U.S.A. Therefore, there can be no doubt that the automobile industry is playing a most important part in earning hard currency for this country.

Throughout the industry it is thoroughly appreciated that there are still difficult times ahead, but there is a quiet confidence that so long as there is reasonable access to overseas markets, British products will successfully continue to compete on both price and quality.

CONTENTS

THE NEW HUMBER SUPER SNIPE

A Completely Redesigned Chassis Powered by the 4·1 Litre Blue Riband Engine

ONE by one, the manufacturers of cars in the larger and more expensive class are abandoning the traditional style and adopting modern lines. The primary reason for this is that with full width bodies, more room can be made available for passengers. No doubt this, together with the fact that the streamlined style is regarded as representing the enlightened modern conception of design, has influenced public opinion in the all important dollar markets to such an extent that there is now little demand for the traditional type of motor car.

The specification laid down for the new Humber Super Snipe called for a frame that was stiff enough to give a feeling of security when travelling at high speeds. In fact, the torsional stiffness of the vehicle, as measured over the wheelbase by the method recommended by M.I.R.A. is 6,000 lb-ft/deg. A genuine 90 m.p.h. was considered to be necessary, in order that all journeys, and especially the long ones, which are commonplace in many overseas countries might be completed in the shortest possible time. Adequate brakes have been incorporated to ensure good control over this car, which weighs 3,871 lb dry; and it is claimed that their recovery after immersion in water, for instance when fording streams, is exceptionally good. Comfort and ease of handling are the other essentials for a vehicle of this type. Large section tyres have been employed, and the heavy duty shock absorbers are suitable for even the roughest terrain in colonial lands. Considerable attention has been given to ensuring that the steering linkage is stiff enough for precise control at high speeds.

SPECIFICATION

ENGINE: Six cylinders. Bore and stroke 3·5 in (88·9 mm) × 4·375 in (111·13 mm). Swept volume 252·6 in³ (4,138·8 cm³). Maximum b.h.p. 116 at 3,600 r.p.m. Maximum b.m.e.p. and torque respectively 126 lb/in² and 211 lb-ft at 1,400 r.p.m. Compression ratio 7·13:1. Fully balanced, seven bearings forged crankshaft. Overhead valves, push rod operated. Stromberg D.B.V.A. 42, downdraught carburettor with 1¼ in choke.

TRANSMISSION: Borg and Beck single dry plate clutch, 10 in diameter. Four forward speeds and one reverse. Ratios: top 1:1, third 1·42:1, second 2·09:1, first 3·12:1, reverse 3·31:1. Propeller shaft, Hardy Spicer open.

REAR AXLE: ¾-floating unit, with hypoid pinion, and a final drive ratio of 3·9:1 or 3·7:1.

FRONT SUSPENSION: Double transverse wishbone link, with coil spring and anti-roll bar, and Woodhead Monroe Monroe-Matic 1¼ in diameter shock absorbers.

REAR SUSPENSION: Semi-elliptic leaf spring with through axle, and Woodhead Monroe Monroe-Matic 1¼ in diameter shock absorbers.

STEERING: Burman worm and nut. Ratio, 19·3:1, giving 4½ turns of wheel from lock to lock. Turning circle, 43 ft 6 in.

BRAKES: Front, Lockheed hydraulic two leading shoe. Rear, Lockheed hydraulic, leading and trailing shoe. Drum diameter, 11 in. Shoe width, 2¼ in. Total friction area 191 in².

TYRES: 7·00 × 15·00 on 5½ in rims. Pressure: front and rear 24 lb/in².

DIMENSIONS: Wheelbase 9 ft 7¾ in. Track: front 4 ft 9·94 in, rear 4 ft 8·25 in. Dry weight 3,871 lb. Ground clearance 7·4 in. Overall length 16 ft 5 in. Overall width 6 ft 1½ in. Overall height 5 ft 6 in.

Engine

The Blue Riband engine is basically the same as the Superpoise, which was described in the May 1952 issue of *Automobile Engineer*. A smaller flywheel and clutch are employed in the Blue Riband unit, and at the front end of the crankshaft a Metalastik torsional vibration damper is fitted. The overall dimensions of the flywheel and ring gear are 15 in diameter by 1¾ in thick, and the weight is 43 lb. On the new unit, the push rods are tubular instead of solid, and a different carburettor is employed. A modified breathing system has been adopted and a smaller fan is carried at the front. The cylinder bores are no longer chromium plated, but chromium plated top rings are employed.

At 3,600 r.p.m. the 4,138 cm³ engine develops 116 b.h.p. This gives 60·08 b.h.p./ton, which is adequate to ensure a lively performance. The b.h.p./in² piston area is 2·02 and the b.h.p./litre is 28·03. A dry weight of 758 lb is quoted for the engine so b.h.p./lb is 0·153. At 1,400 r.p.m. the maximum b.m.e.p. is 126 lb/in².

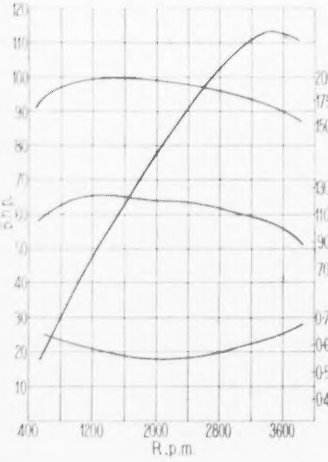
At 3,600 r.p.m. the mean piston speed is 2,625 ft/min. The stroke : bore ratio is 1·25 : 1, and the connecting rod length : stroke ratio is 1·885 : 1. A minimum brake specific fuel consumption of 0·55 pt/b.h.p.-hr is obtained. The overall dimensions of the engine, less air cleaner and flywheel, are 29¾ in high by 22 in wide by 37¾ in long.

A somewhat unusual engine mounting arrangement has been adopted.

The modern style of the Humber Super Snipe gives ample interior width

All the mounting units are supplied by the Firestone Tyre and Rubber Company Ltd. A conventional V-arrangement of two rubber sandwich units is used at the front. The rubbers are each interposed between two brackets one bolted to the side frame and the other to the crankcase.

At the rear, the mounting is carried on a plate held by the bolts securing the rear cover to the gearbox. The upper and lower edges of this plate are flanged rearwards. Above the rear extension of the gearbox, the upper flange forms part of another V-type rubber sandwich mounting. The two rubbers are interposed between the flanges and brackets bolted to a cross-member attached to the front arms of the cruciform. They are above the axis of oscillation and are focused on it. However, since the distance between the axis of oscillation and the centre of the universal joint is large, it has been necessary to incorporate another rubber unit to prevent undue lateral oscillation of the propeller shaft. This unit consists of a steel T-piece welded to the lower flange, which is under the rear extension, and two rubber blocks mounted, one each side of the leg of the T-piece, on a cross-member between the two front arms of the cruciform bracing. The rear end of an adjustable engine tie rod is bolted under this cross-member; the front end of the rod is carried in a rubber bush between two lugs cast on the bottom of the bell housing.



Engine performance curves with dynamo and air cleaner fitted

CAMSHAFT PERFORMANCE DATA AT 3,600 R.P.M.

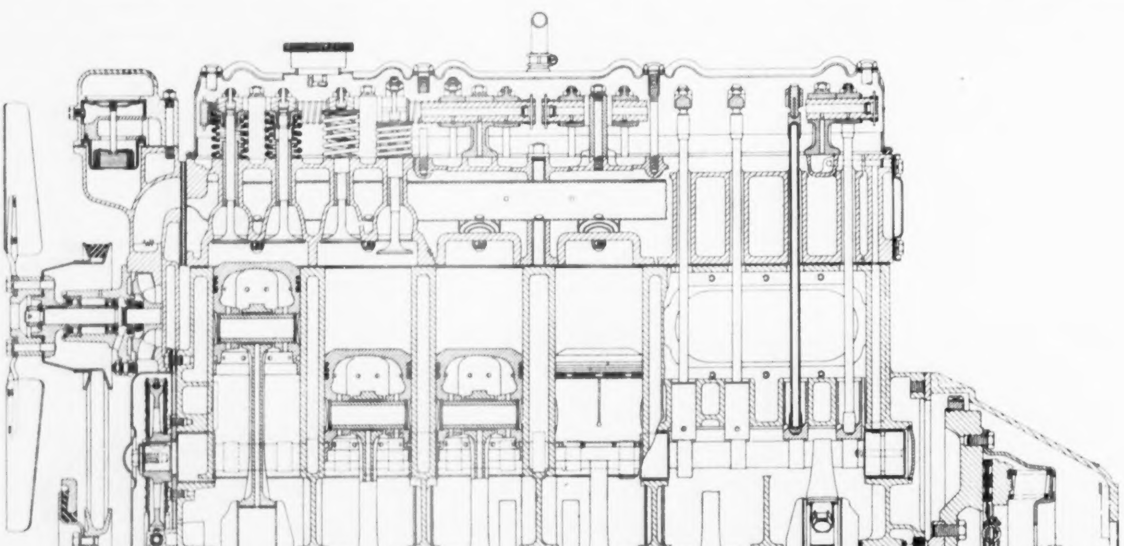
Maximum tappet positive acceleration (flank)	6,460 ft/sec²
Maximum tappet negative acceleration (nose)	2,210 ft/sec²
Maximum tappet velocity	7·61 ft/sec
Lift at cam	0·3290 in
Nominal period of cam	126 deg

A Stromberg DBVA 42 carburettor is mounted on a block of asbestos based material on top of the manifold. The choke diameter is 1¼ in. A 0·065 in main jet is fitted together with a 0·056 in by-pass jet. The idle discharge holes are 68—60, and the high speed bleed is 52. The main discharge tube is L-2080. An A.C. mechanical fuel pump actuated by an eccentric on the camshaft delivers the fuel through a Fram fuel filter to the carburettor. The fuel tank is bolted up to the boot floor. Its capacity is 15 gallons.

A conventional rotor type water pump delivers the coolant into a distributor tube in the cylinder head whence it is directed around the exhaust valve seats and sparking plug bosses. A subsidiary thermo-syphon effect is employed to cool the cylinder walls. The water outlet is through a thermostat valve, which lifts at 169 deg F. A Serck radiator with five rows of copper gilled brass tubes is employed. Its frontal area is 339·1 in², and the block thickness is 2¾ in. A valve in the filler cap maintains a pressure of 4 lb/in².

Clutch and gearbox

A 10 in diameter Borg and Beck 10AG-G single dry plate clutch is employed. The friction area is 42¾ in² per side. Twelve pressure springs are incorporated and they give an assembled load of 120 to

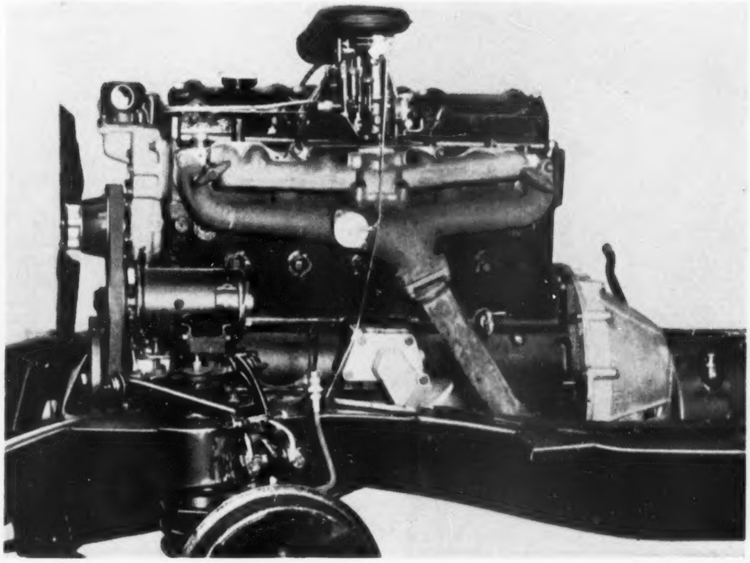

The Blue Riband engine is basically the same as the Commer Superpoise unit

130 lb. Clutch actuation is through a graphite thrust ring. The clutch bell housing is made of aluminium.

A BS 1452 Grade 10 cast iron gearbox casing is employed. The rear extension and side cover are DTD 424 aluminium die castings. When dry, the whole unit weighs 118 lb, and the oil capacity is 5 pints. The gear ratios are: top 1 : 1, third 1·42 : 1, second 2·09 : 1, first 3·12 : 1 and reverse 3·31 : 1. All four forward speeds are engaged through a synchromesh mechanism, and a steering column gear shift control is employed.

The front end of the En 18C primary shaft is $\frac{3}{4}$ in diameter where it is carried in a sealed ball bearing in the tail end of the crankshaft. Immediately behind the driving splines it is $\frac{7}{8}$ in diameter. Towards the rear, it is shouldered and is surrounded by an Angus or Gaco oil seal carried in an aluminium die cast housing bolted to the front wall of the gearbox.

Immediately behind this seal the shaft is shouldered to $1\frac{9}{16}$ in diameter where it is mounted in a single row, ball bearing in a circular aperture in the front wall. A drain hole through the front wall communicates with the space between the seal housing and the bearing. The inner race of the bearing is located between a large diameter washer against the front face of the primary gear, and a chip shield and a circlip in a groove round the shaft. The outer race is located against rearward motion by a circlip in a groove round its periphery. This circlip is retained between the oil seal housing and the gearbox wall. Forward motion of the race is prevented by the oil seal housing, which is pulled up against its front face.



The left-hand side of the engine installation

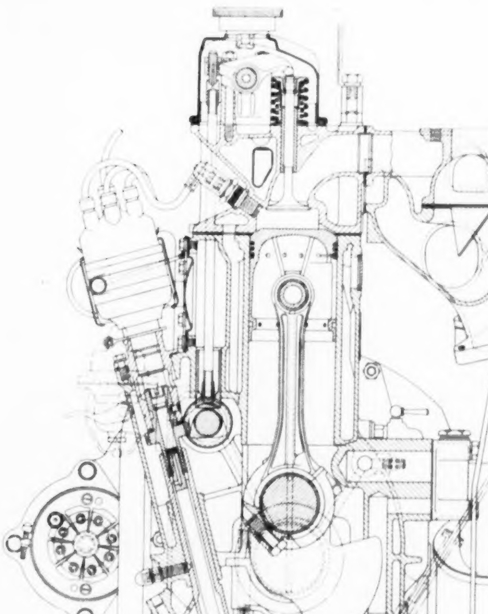
The primary gear is up-ended on the shaft and incorporates the cone seating and splines for engaging the sleeve of the synchromesh mechanism. The included angle of the cone is 15 deg. A counterbore in the centre of the primary gear houses the roller bearing carrying the mainshaft, and radial drillings feed oil from between the gear teeth into the counterbore and thence to the bearing.

The front end of the En 351 mainshaft is reduced to a diameter of $1\frac{1}{32}$ in to form the inner race of the $\frac{1}{2}$ in long roller bearing that supports it in the counterbored primary shaft. The outer race is formed by the counterbore, in which is a groove for a circlip retaining the 21 rollers. Immediately behind this bearing, the mainshaft is $1\frac{5}{8}$ in diameter and splined to carry the En 3A centre member of the Borg Warner type synchromesh unit. A circlip in a groove round the front end of the splines retains this member.

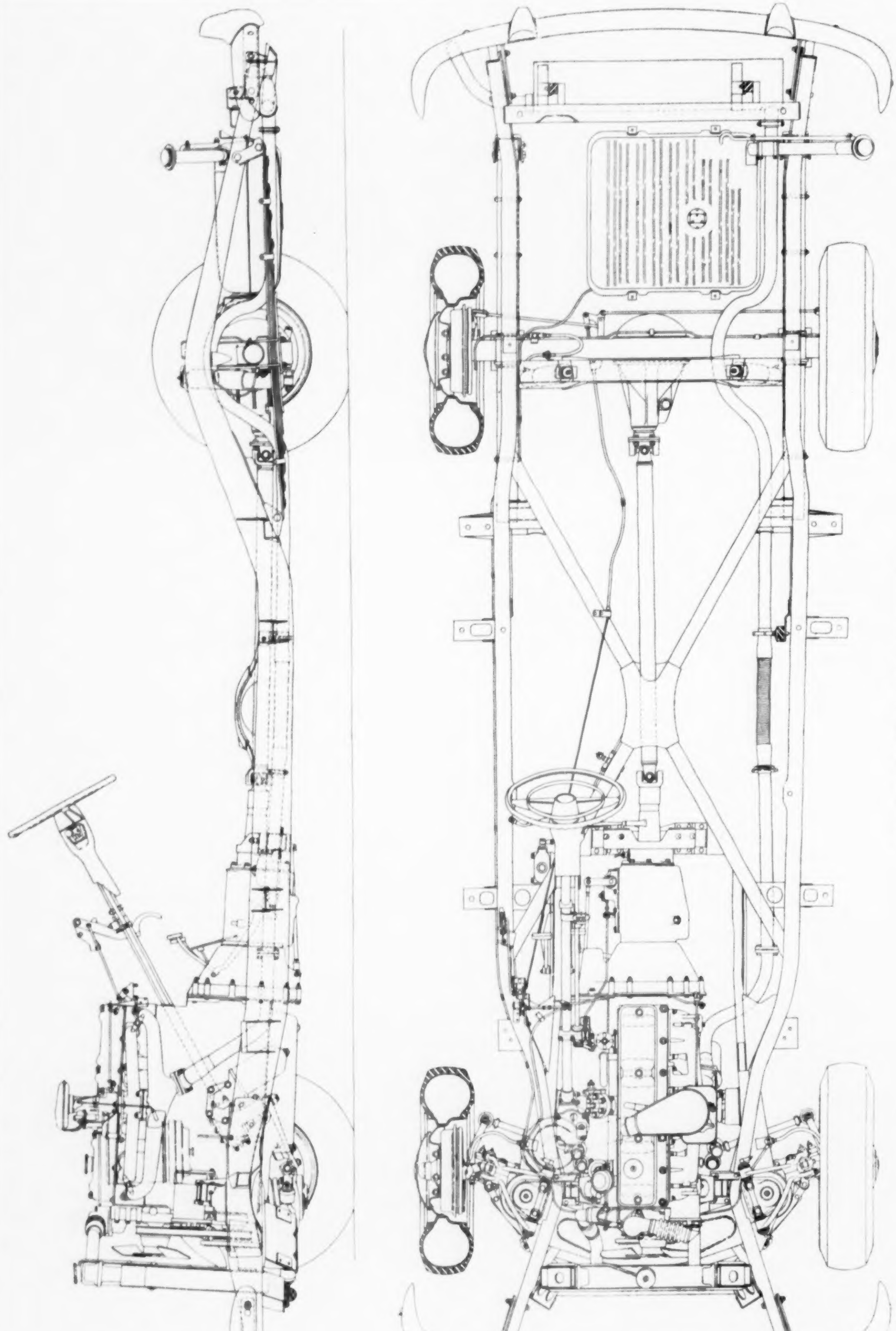
An En 18C sleeve member is employed. It is splined on to the flanged centre member, which is slotted radially to carry three pressed steel channel section struts. The axes of these struts are parallel to the axis of the shaft, and each has a detent pressed in its centre. These detents engage in a groove around the inner periphery of the sleeve. They are held there by the radial pressure exerted outwards by two spring rings. These rings are housed in recesses formed between the flange and the boss at each end of the centre member. They are retained in position by the cone rings. One end of each of the rings is turned into one of the channel section struts to prevent the ring from rotating; thus eliminating any possibility that a strut might become jammed in the ring gap.

The DTD 147A aluminium bronze baulking rings are shouldered, and the small diameter portion is carried in the recess in the centre member. Clearance slots are machined in the rings to receive the ends of the struts. This arrangement allows the rings a restricted rotational movement relative to the centre member. The large diameter portion of each ring is splined and is clear of the sleeve member in the neutral position.

When the sleeve is moved axially to engage a gear, the struts push the appropriate baulking ring before them until it lightly engages the cone. If the rotational speeds of the driving and driven portions are different, the ring rotates a few degrees relative to the sleeve and centre member until the



The bore and stroke are $3\frac{1}{2}$ in and $4\frac{3}{8}$ in respectively



CHASSIS GENERAL ARRANGEMENT

Wheelbase : 9 ft 7¾ in. Track : Front 4 ft 9·94 in ; Rear 4 ft 8¼ in

clearance between the struts and slots is taken up. This rotation sets the splines on the baulking ring out of alignment with those in the sleeve. Thus, further movement of the sleeve, which slips the detents out of their groove, forces the baulking ring harder on the cone to effect the synchronization.

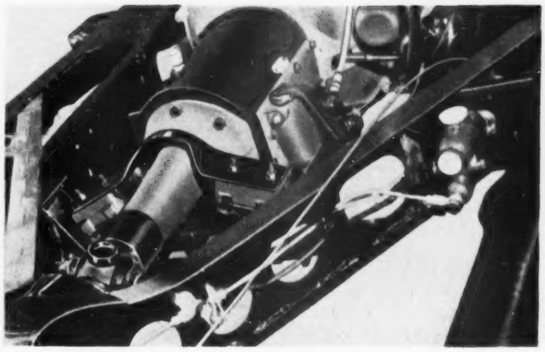
Both on the sleeve and on the ring, the ends of the splines are chamfered at 120 deg included angle, so that when the speeds are synchronized, the sleeve rides over the chamfer and through the splines on the ring and engages the splines on the appropriate gear, thus connecting the driving and the driven members.

The ends of the splines on the gears are also chamfered at an included angle of 120 deg to provide a lead-in for the sleeve splines. Slight rotation of the baulking ring may be necessary before this final motion can be completed. In order that the ring may disengage and rotate easily, and also to enable it to penetrate the oil film on engagement, a spiral groove is cut around its conical face.

All gears are of En 18C. With the exception of the reverse gear train, they have a diametral pitch of 8, and a helix angle of 28 deg. Straight spur gears are employed for the reverse train which are the only gears that are not in constant mesh. Their diametral pitch is 8/10.

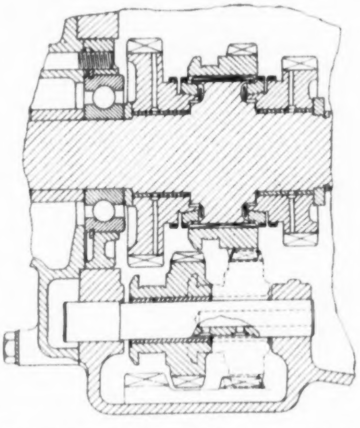
The third speed gear is carried on a Clevite, rolled strip type bush round the 1⅝ in diameter portion of the main shaft. This bush is pressed into the gear, and 3/16 in diameter oil holes are drilled in it to communicate with an annular recess in the gear. Four radial holes from the roots of the teeth communicate with this recess, and the bush is lubricated by oil forced through the holes by the engagement of the teeth.

The synchro-cones and spline collar are integral with the gear. Axial movement is restricted at one end by the centre member of the synchromesh unit and at the other by a ¼ in thick, En 2B thrust washer, which is retained against a shoulder on the shaft by a circlip. The rear end face of the gear is counterbored about 5/32 in to clear this circlip.



Below the rear engine mounting is a steadying arrangement comprising a T-piece between two rubber blocks

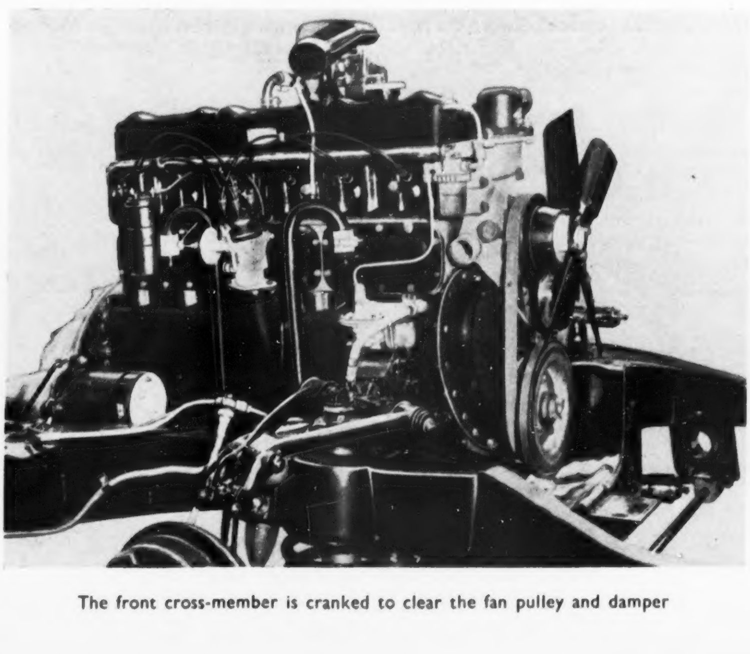
The arrangement of the second speed gear is identical with that of the third speed one except that it is carried on a 1 13/16 in diameter portion of the shaft and the synchro-cone is at the rear instead of in front. The only other difference is that this gear has no counterbore at its front end where it bears against a thrust washer, since there is no circlip to be cleared. However, radial grooves are cut in its end face to provide for the lubrication of the surfaces taking the thrust. The washer itself is prevented from rotating by a dowel which registers in a slot cut in its inner periphery, and which is carried in a radially drilled hole in the shaft.



The reverse idler spindle is stepped to prevent its being assembled the wrong way round

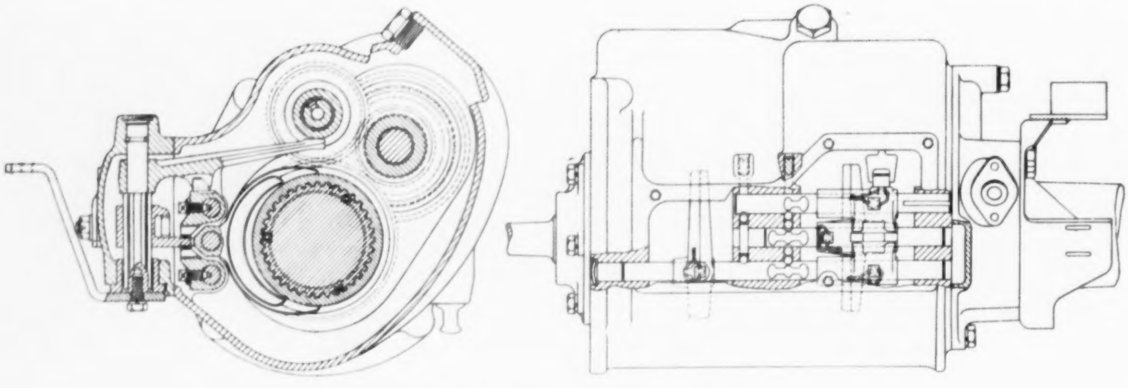
Unlike the front synchro unit, the centre member of this one is integral with the shaft, and the reverse gear is machined on the sleeve member. The cone angles are the same as at the front, and the unit is similar in every other respect. The first speed gear, which again is integral with the cone and spline collar, is located between the centre member of the synchro unit and a 0·203 in thick thrust washer of En 32B. This gear is also carried on a rolled strip type bush lubricated in the same way as the others. Grooves are incorporated in its end face for the lubrication of the bearing surfaces between it and the thrust washer.

To the rear of the thrust washer are the inner race of the ball bearing supporting the main shaft in the casing, a distance piece about 1 3/16 in long, the En 32A speedometer gear, a tab washer and a ring nut. This nut pulls the whole assembly against a shoulder on the shaft immediately behind the first speed gear. Behind the nut, the shaft is waisted to 1 in diameter and at the extreme rear are cut the splines for the sliding joint. The overall diameter of the splines is 1·375 in, and the root diameter is 1·137 in.



The front cross-member is cranked to clear the fan pulley and damper

The outer race of the ball bearing immediately behind the first speed gear and thrust washer is carried in a circular housing, the diameter of which is about 1/16 in greater than that of the largest of the mainshaft gears. This housing is secured to the front face of the DTD 424 rear extension casting by four ⅜ in diameter bolts. These bolts also secure the bracket for the rear mounting units. The rear extension casting bears against the outer race to prevent it moving to the rear,



The selector rods are carried in the side of the gearbox

and a circlip in a groove round the race bears against the housing to prevent it moving forwards. The whole mainshaft gear assembly, mounted on the extension, may be inserted from the rear. It is located axially by the set bolts securing the extension to the rear wall of the gearbox. Radial location is, of course, effected by the register of bearing housing in the hole in which it is carried in the rear wall.

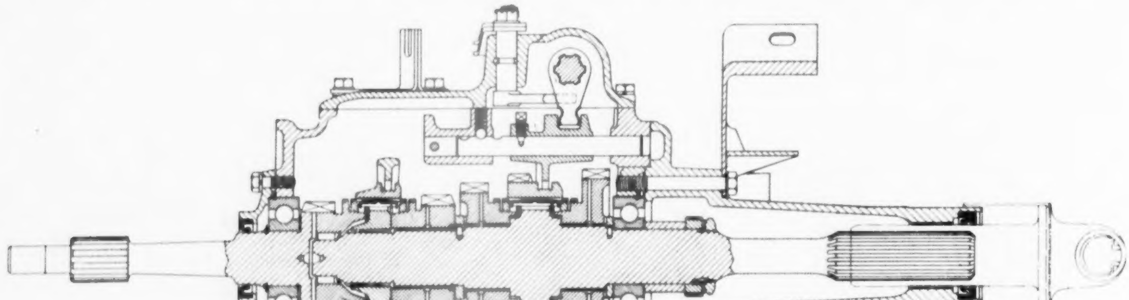
All five layshaft gears are integral with the sleeve which is made of En 18C. This sleeve is supported at each end by needle rollers, 13/16 in long, on the 1 in diameter En 32B shaft. Thirty rollers are employed in each bearing, the outside diameter of which is 1¼ in. Two Clevite thrust washers are fitted, one at each end, and each is located against rotation by pressing a small portion of its periphery into a slot in the casing. The thrust washer at the front is between the sleeve and the casing. At the rear, a floating thrust washer of En 26 is interposed between the fixed thrust washer and the casing. This steel washer controls the end float. The shaft is a press fit in the casing. Positive axial location is provided for both the layshaft and the reverse idler spindle by a plate, which is engaged in slots cut chordwise near their ends, and which is bolted to the rear wall of the gearbox between the two.

GEARBOX DATA

	Pitch Circle diameter	Helix angle	Diametral pitch
Primary gear	2·875 in	28 deg	8
3rd speed gear	3·50 in	28 deg	8
2nd speed gear	4·125 in	28 deg	8
1st speed gear	4·875 in	28 deg	8
Layshaft cluster :			
Layshaft gear	4·375 in	28 deg	8
3rd speed gear	3·75 in	28 deg	8
2nd speed gear	3·00 in	28 deg	8
1st speed gear	2.375 in	28 deg	8
Reverse gear	2·125 in	—	8/10
Reverse idler	2·875 in	—	8/10

The En 43B reverse idler spindle is 0·868 in diameter, except at its rear end which is slightly larger for a length of about 1¼ in where it is carried in the gearbox wall. This arrangement has been adopted to ensure that assembly the wrong way round is impossible. The front end of the spindle is in a boss on the side wall, and is drilled axially for about 2¾ in. Two radial drillings from the axial hole supply oil to a longitudinal slot in the periphery. This slot is covered by a Clevite bush pressed in the En 18C reverse gear when in the engaged position. It is in order that the lubrication system shall function properly that it is essential for the spindle to be assembled the correct way round.

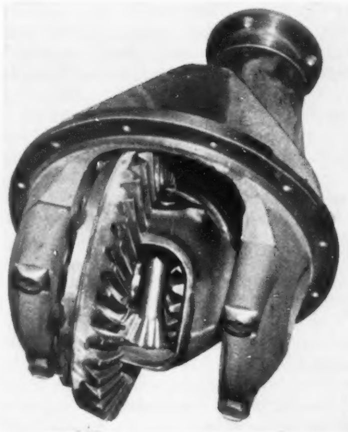
The selector and striker arrangement is somewhat different from that adopted



Borg Warner type synchromesh units are employed for all four forward speeds

by most other manufacturers. The three selector rods are positioned horizontally, one above the other, in bosses on the right-hand side of the gearbox. A striker lever is carried on a vertically splined spindle to the right of the rods; it is slid vertically on the splines to engage the appropriate striker collar. The sliding motion is effected by means of the selector lever the end of which engages in a fork on the striker lever boss. This selector lever is mounted on a horizontal spindle positioned with its axis transversely immediately in front of the vertical striker spindle. Both spindles are carried in the DTD 424 side cover of the gearbox.

The operating lever on top of the striker spindle is made from $\frac{1}{4}$ in thick plate. Its outer end is drilled and connected to the linkwork from the steering column gear shift mechanism, and its inner end is carried in a slot cut diametrically across the upper end of the splined boss. This boss is mounted on top of the striker spindle. It is flanged at its upper end and bears in a hole in the top of the cover. A $\frac{5}{16}$ in diameter set bolt secures the lever to the spindle. This bolt is passed through a hole in the pivot end of the lever and screwed into an axially drilled, tapped hole in the spindle. The lower end of the spindle is turned down to $\frac{5}{8}$ in diameter and carried in a boss in the base of the cover. Axial location



The two pinion hypoid gear carrier assembly

is effected by a pin driven through the boss and engaging in a groove round the spindle.

Selection of the forward gears is made by moving the En 32A drop forged striker lever vertically on its splines. This motion is effected by the En 5 selector lever, the end of which engages in a fork formed in the splined boss of the striker lever. One arm of the fork is extended to engage in the forks that are integral with the collars carried on the selector rods. The En 8 selector spindle is mounted in a $1\frac{5}{8}$ in long, drilled boss in the side of the cover. The outer end is shouldered and threaded, and the operating lever, to which is connected the cable from the steering column control, is clamped against the shoulder by a nut on the threaded end. The selector lever is forged integrally with the spindle. Axial location is effected by a pin engaging in a groove round the centre of the spindle.

The $\frac{9}{16}$ in diameter En 43B selector rods for first and second gears and reverse are $5\frac{1}{2}$ in and $6\frac{1}{2}$ in long respectively, while that for third and top is about $9\frac{3}{4}$ in long. Reverse gear is engaged by the lower rod on which is a collar that carries a conical ended, square headed screw engaging in a tapered hole in the rod to provide location against rotational and axial movement. The forks in which the end of the striker lever is engaged are formed on top of this collar, and a tongue extending below it registers in a fork on the reverse idler selector lever. This lever, which is made of En 32, is pivoted on the plain portion below the splines of the striker spindle, and its outer end engages in a channel round the reverse idler gear boss. To prevent rotation of the selector rod, and, consequent disengagement of the various forks and levers, a slot is cut

A cable control is employed to operate the selector lever, and a rod for the striker control

longitudinally in the periphery at its rear end, and in it is registered the dowel end of a grub screw in the side cover.

A drop forged En 5 selector fork, together with the striker fork, is integral with the collar assembled on the centre rod and secured by a conical headed screw, similar to that used for the reverse collar. On the top rod, the selector fork and striker forks are on separate collars which are also secured by conical headed screws. Spring loaded balls in horizontal drillings in the bosses carrying the front ends of the rods engage in grooves to form the gear locks.

The gear interlock, between the reverse and first and second, and between the first and second and the third and top gear selector rods, is formed by balls in a vertical drilling, two between each pair of rods. The interlock between the third and top, and reverse selector rods is formed by two balls, one on each side of a plunger in a vertical drilling between the two rods. Only one gear can be selected at a time because movement of one of the rods forces the balls out of the grooves in its periphery and seats the adjacent balls in grooves in the periphery of the other two rods, which cannot then be moved.

In the rear extension of the gearbox, the sliding joint sleeve is carried in a Clevite 10 bush. An Angus oil seal is carried in the extreme rear end of the extension, and an oil drain hole is drilled in the housing for the ball bearing carrying the main shaft in the rear wall of the gearbox. A sleeve carried on the flange of the universal joint enshrouds the seal housing and screens it from foreign matter.

Back axle

A $2\frac{1}{4}$ in diameter Hardy Spicer open propeller shaft, 46 in long, transmits the drive to the three-quarter floating hypoid rear axle, which is housed in a banjo type casing. The oil capacity is 4 pints, and EP 90 gear oil is recommended. An oil filler plug is incorporated in the nose piece casting and the drain plug is in a boss welded to the $\frac{1}{8}$ in thick pressed steel rear cover, which is welded to the banjo casing. The final drive ratio is 3·9 : 1, or an alternative ratio of 3·7:1 may be obtained by fitting a different crown wheel.

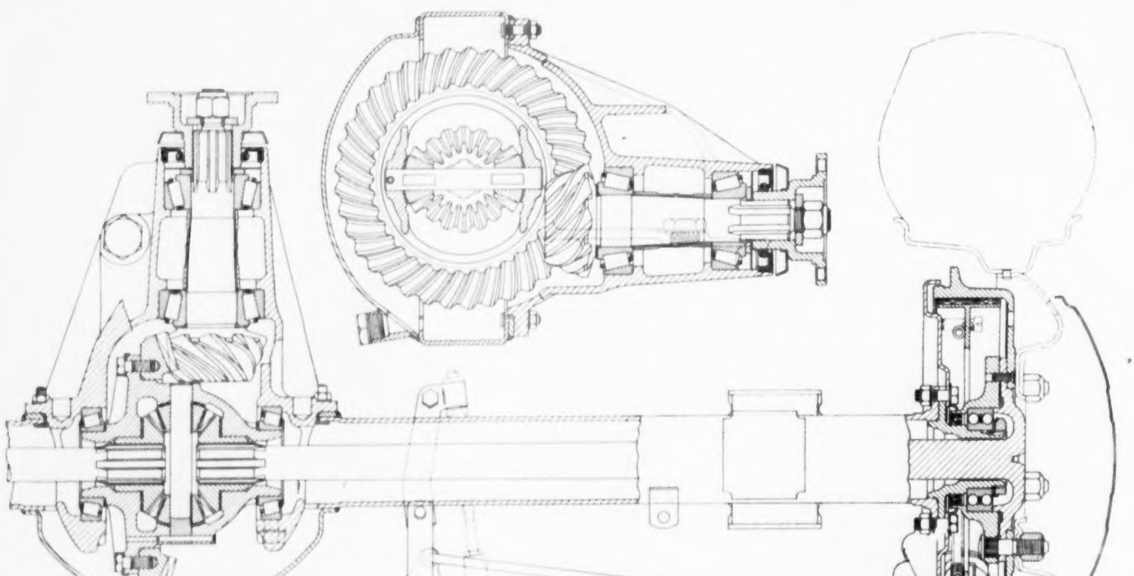
All the gears are carried in the BS 1452 Grade 14 cast iron nose piece which is secured to the banjo casing by twelve $\frac{5}{16}$ in diameter special bolts. These bolts are knurled for a length of about $\frac{1}{4}$ in immediately beneath their heads. They are a press fit in the casing. This arrangement has been found preferable to using studs because the bolt head beds down on the inside of the casing and obviates the possibility of oil leaks.

The hypoid pinion is integral with its En 352L shaft, and is overhung from two taper roller bearings. The axis of the pinion shaft is offset $1\frac{1}{32}$ in to the right of the axis of the differential pinion spindle and $1\frac{1}{2}$ in below the axis of the crown wheel. The companion flange for the universal joint is splined on to the front end of the shaft and secured in the usual manner by a self-locking nut and a $\frac{1}{4}$ in thick washer. An Angus M3026 oil seal is carried in the front end of the nose piece. It bears on the boss of the universal joint companion flange. A dished steel shroud is pressed on to the boss to protect the seal from foreign matter.

Assembled on to the shaft in the following order are: shims, which are interposed between the inner race of the rearmost roller bearing and the pinion, to control the meshing of the crown wheel and pinion; a $2\frac{3}{4}$ in long En 8 swaged distance tube; the front roller bearing; a thick distance washer; and an oil thrower disc which helps to shield the oil seal from excessive oil splash. This whole assembly, together with the universal joint flange, is pulled up by the nut on the end of the shaft. This nut is tightened until the pre-load on the bearings is such that a 14-17 lb-in torque is required to turn the shaft. The outer races of the bearings are pulled up against shoulders in the casing.

An En 352 crown wheel is employed. To obtain the larger ratio 39 teeth are incorporated, and its diameter is approximately 9·103 in. When the smaller ratio is required, a 9·108 in diameter crown wheel with 37 teeth is employed. Eight $\frac{3}{8}$ in diameter set bolts are passed through a flange on the one piece malleable cast iron differential cage and screwed into tapped holes in the crown wheel to pull it up to the inner face of the flange. Between each pair of bolts is a radial stiffening rib.

The two $2\frac{5}{16}$ in diameter En 352 differential pinions are carried on a common spindle, which is $\frac{3}{4}$ in diameter and made of En 352. Axial location of the spindle is effected by a pin driven through a hole in the cage and the end of the spindle. The hole in the spindle is countersunk at each end to provide a lead-in for the pin. To ensure that the journal bearing surfaces between the pinions and the spindle are adequately lubricated, flats are machined on the surface of the spindle beneath



Alternative final drive ratios of 3·9 : 1 or 3·7 : 1 are obtained by fitting different crown wheels

the gears. These flats also feed oil from the centre out to the spherical thrust washers interposed between the pinion and the cage. The thrust washers are $1\frac{5}{8}$ in outside diameter and they are made of STA/7/CPI bronze.

The En 352 differential gears are approximately 3·755 in outside diameter, and the effective length of their tooth engagement with the pinions is $\frac{3}{4}$ in. The back lash is 0·006 to 0·008 in. Flat thrust washers of STA/7/CPI bronze are interposed between the gears and the cage. These thrust washers and the spherical ones for the differential pinions have holes drilled in them to retain lubricant. The journal bearing length of the gear boss in the differential cage is $\frac{7}{8}$ in, and its diameter is $1\frac{7}{8}$ in.

Two taper roller bearings, spaced with their centres $5\frac{1}{4}$ in apart, support the cage. Pre-loading the bearings is effected by springing their housings to accommodate the bearing and cage assembly, and it is controlled by shims between the inner races and shoulders on the cage. These shims are also used to control the meshing of the crown wheel and pinion gears. Tooling holes are drilled in the outer face of the casing immediately in front of the bearing; these are used to spread the housing when inserting the bearing and differential cage assembly.

Although no special measures are taken to provide lateral support for the bearing caps these, as can be seen from the illustration, are of exceptionally stiff cross-section, since their outer faces incorporate a relatively large flange. This flange is backed up by a semi-circular rib round the outer face of the bearing housing in the nose piece. Thus, when the cap is assembled to the nose piece, the flange and rib together form a complete circular rib immediately outboard of the bearing. Furthermore, on the outside of the casing, gussets approximately $\frac{5}{16}$ in thick are extended forward on each side, from a point immediately in front of the bearing cap bolts, to the front of the nose piece. Apart from these four gussets, there is an additional one between the two on the same side as the crown wheel. This one provides extra support behind the centre of the bearing.

The splined inner end of each En 19C half shaft is passed through a $1\frac{1}{2}$ in diameter hole in the cage and carried in the differential gear. The hole in the cage is countersunk at its outer end to form a lead-in for the shaft. Positive axial location is provided at the wheel bearing, so hardened plugs are not used between the two half shafts. The overall diameter of each shaft at the splines is 1·3725 in, and the root diameter is 1·16 in. Immediately outboard of the splines, the shaft diameter is $1\frac{3}{16}$ in, and it is tapered up to $1\frac{1}{2}$ in diameter at the outer end.

The axle casing is divided at the centre to form the banjo housing, and it is made from 3 in diameter by 9 s.w.g. En 2C tube. An En 3 flange is butt welded on the outer end of the axle tube. Bolted to it are the En 2A brake back plate and the flange of the En 5P hub. A two-row ball thrust bearing is carried on the $1\frac{21}{32}$ in diameter outer end of the hub, and its inner race is pulled up against a shoulder by a ring nut. This nut is locked by a tab washer. The outer race is carried in a cast iron housing, in the inner end of which is an Angus MIS 26 oil seal. The end of the seal housing projects into a shroud ring secured to the brake back plate by the six $\frac{3}{8}$ in diameter bolts attaching it to the flange on the axle tube. A drainage hole is drilled through the flange to allow any oil or grease that may have collected in the shroud ring to pass to the outside of the brake unit.

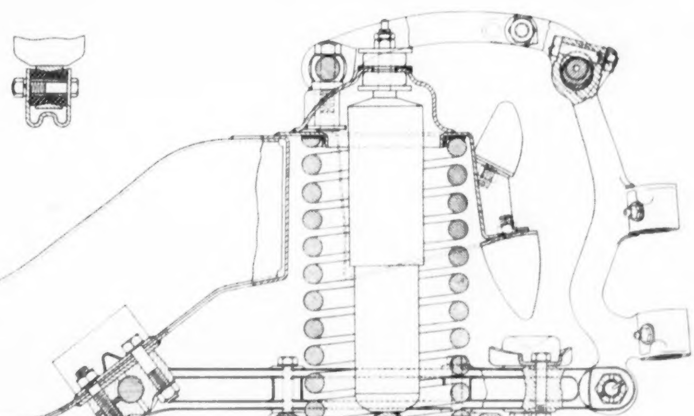
The housing for the outer race of the bearing is flanged, and the upset driving flange on the half shaft is spigoted in to it. The two are held together in the first place by the countersunk set screws that hold the cast iron brake drum on a spigot on the outer face of the driving flange, and secondly by the set bolts and nuts securing the wheel. A rubber sealing ring is carried in a groove round the inner spigot of the driving flange. This ring is pressed between the driving flange and the bearing housing and prevents oil or grease from leaking between the two faces and being flung outwards into the brake drum. It is necessary because the inner end of the spigot bears against the outer race and pulls it up against a shoulder in the housing, so there is a small clearance between the two flanges.

Rear suspension

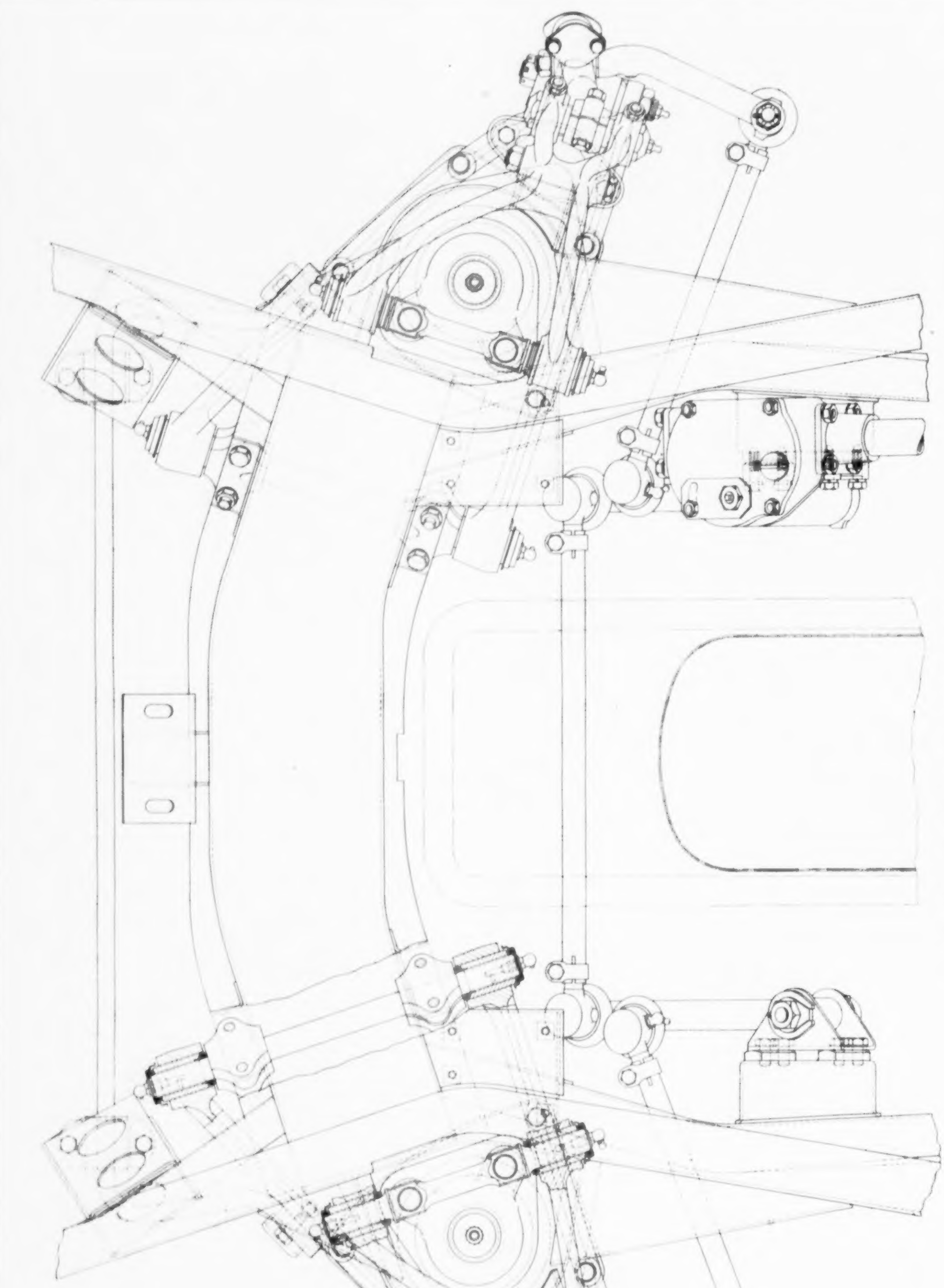
At the rear, semi-elliptic springs are used in conjunction with Monroe-Matic $1\frac{3}{8}$ in diameter telescopic shock absorbers. The lower end of the shock absorber has a rubber bushed ring type end fitting and it is overhung-mounted on a flange turned down at the front of the plate used in conjunction with the U-bolts to hold the spring up under the axle. A rubber sandwich type end fitting is used at the top end of the shock absorber, and it is carried on a frame cross-member about 3 in forward of the axle. It has been necessary to incline the shock absorbers inwards in order to accommodate them in the available space.

Rubber bump stops are fitted on the frame side members and they bear on the axle tube when it comes up to the full bump position. The rubbers are semi-ellipsoidal in shape to give a progressive stopping action. A Balata strap is used for the rebound stop. Its lower end is bolted to the front face of the spring seat bracket welded under the axle tube. A square washer is fitted under the head of the bolt and two square grip washers, in which are pressed perforations with jagged edges, are placed one on each side of the strap. One edge of the plain washer is lipped, and it is positioned in such a way that when the lip bears on the strap there is no danger of chafing, as there would be if a sharp edge bore on it. At the top end, the strap is bolted underneath the frame side member. Similar perforated washers are used here, but because the strap is turned through a right angle at this point, it has been necessary to fit under the bolt head a $\frac{1}{4}$ in thick stamping with a radiused edge instead of the lipped plain washer which is employed at the bottom.

The En 45 spring is 52 in long between the eye centres, and 3 in wide. A $\frac{1}{4}$ in thick top leaf is employed and the bottom one is $\frac{5}{16}$ in thick. The other six are of 7BWG strip. Rubber interleaving pads are fitted between the ends of the top five leaves only. The front eye is carried between lugs formed by a $\frac{1}{4}$ in thick U-bracket welded and riveted under the frame side member, and the rear one on a $\frac{1}{4}$ in thick U-section shackle. At the top end, the base of the U-section of the shackle is



The position of the bump and rebound stops is such that they will not induce heavy bending stresses in the wishbones



FRONT SUSPENSION AND STEERING LAYOUT

An 18 deg angle of trail has been incorporated in the front suspension units so that the front cross-member may clear the sump

cut away so as to leave two arms which are positioned one each side of the frame side member. They are carried on a bolt passed transversely through a bush in the member. This bolt and the others in the spring eyes are of En 8Q and they are $\frac{9}{16}$ in diameter, Metalastik rubber bushes are employed in the spring eyes and side members.

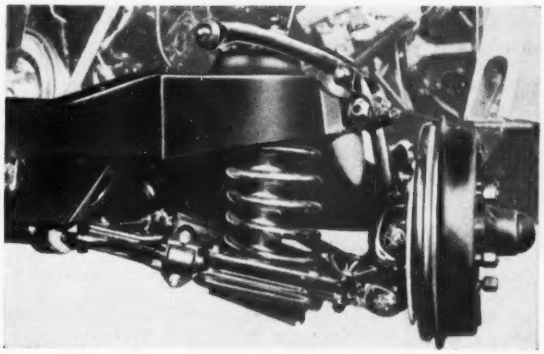
In order to give a slight under-steering tendency, the axle is offset $2\frac{7}{8}$ in forward of the centre of the spring. The unsprung weight is 432 lb. To the fully laden position the deflection is 9·26 in and to full bump 12·01 in. With a total rate for both springs of 216-300 lb/in, the periodicity is 61·6 cycles/min.

Front suspension

A double transverse wishbone and coil spring arrangement is used at each side on the front suspension. The principal angular settings are: swivel pin angle 5-$5\frac{1}{2}$ deg, castor angle $1 \pm \frac{1}{4}$ deg, camber angle $\frac{1}{2}$-1 deg. The toe-in is $\frac{1}{8}$ in. Monroe-Matic $1\frac{1}{2}$ in bore telescopic shock absorbers are mounted co-axially with the springs.

The distance measured perpendicularly from the axis of the top inner wishbone bearing to the centre line of the chassis is $16\frac{1}{2}$ in, while that from the lower bearing is $10\frac{5}{16}$ in. Between these two bearings the vertical spacing is 11·06 in, and the vertical spacing between the outer two bearings is 10·96 in. The length of the upper link is $8\frac{3}{4}$ in and that of the lower one 16·92 in. The angle of trail is 18 deg.

Rubber bump and rebound stops are bolted to the outer edge of the upper spring pan. The upper rubber is rather low relative to the pivot of the top wishbone arm against which it bears in the rebound position. Moreover, its radial distance from the pivot is only $6\frac{3}{4}$ in. Accordingly, instead of being truly semi-ellipsoidal as is the bump stop, its axis is curved to conform more or less to the arc of the circle in which the area on which it contacts on the wishbone arm moves. This arrangement has been adopted in order to prevent undue strain being imposed on the rubber by the scrubbing action that would otherwise occur.



The independent suspension unit viewed from the front

The unsprung weight is 133 lb per side. When the vehicle is laden, the wheel deflection is 4·20 in, and to full bump it is a further 2·85 in. The front end rate is 68·3 lb/in and this gives a periodicity of 65 cycles per minute. This is obtained with a spring whose rate is 240 lb/in. The spring is made from 0·67 in diameter En 45 grade 2 ground rods, hot coiled, having a Rockwell C hardness of 40-47. Nine coils are incorporated and the overall diameter is 5·9 in. In the free condition, the spring length is 19·03 in, and when fitted it is 10·70 in. Rubber seating rings, about $\frac{1}{8}$ in thick, are used at each end of the spring.

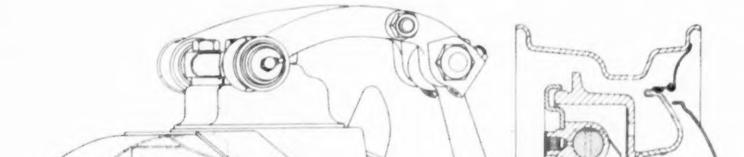
A two piece En 18T stamping forms the upper wishbone link. The two parts are clamped together near the outer end by a $\frac{1}{2}$ in diameter bolt, and location is effected by a $\frac{1}{2}$ in diameter dowel just inboard of the bolt. The inner bearings are carried on an En 32M fulcrum pin forging. This forging is pulled down by two $\frac{1}{2}$ in diameter set bolts on to two platforms pressed in a dome welded above the spring pan. The bolts are screwed into slugs welded under the platforms. The dome also carries the sandwich type rubber mounting at the top of the shock absorber.

The ends of the fulcrum pin forging are turned down to $\frac{7}{8}$ in diameter and are threaded $\frac{7}{8}$ in Whitworth to take the En 1A bushes pressed into the bosses at the two ends of the wishbone. The effective bearing length of these bushes is $1\frac{3}{8}$ in and the bearing centres are spaced $8\frac{1}{4}$ in apart. Rubber sealing washers and pressed steel retainers are fitted at both ends of each bush. A large fillet radius, about $\frac{7}{32}$ in, is incorporated at the inner end of each of the threaded portions of the pin to avoid stress concentrations, and the plain steel retainer ring rides on the fillet. The retainer at the outer end is dished to surround the projecting end of the bush so as to protect it from foreign matter. Lubrication is effected through grease nipples screwed into the ends of the pin.

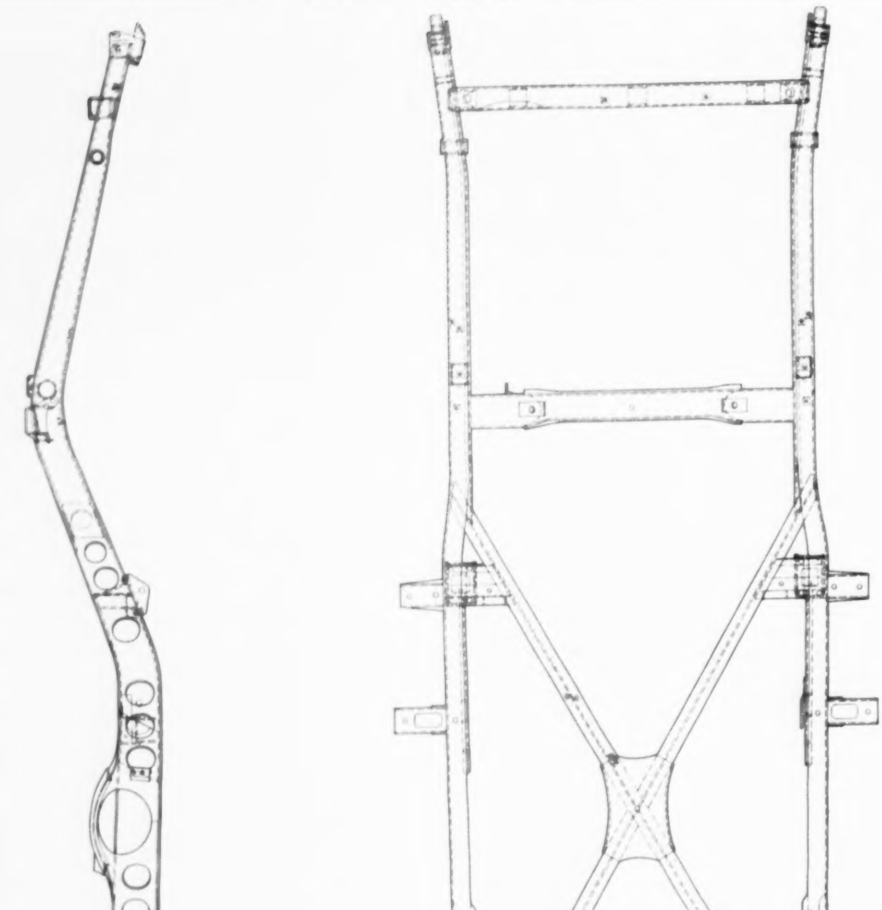
At the outer end, an En 1A pin is carried in screwed bushes in two lugs, one on each half of the wishbone. The bushes have thimble ends, and they are assembled one on each end of the pin. A grease nipple is screwed in the end of the rear one, and a hole is drilled axially through the pin to allow the grease to pass from the rear to the front. Each bush is secured by a cotter pin in the lug in which it is carried.

An eccentric for camber angle adjustment is formed on the pin between the two screwed bushes. It is carried in a split boss at the upper end of the stub axle carrier forging. A $\frac{5}{8}$ in diameter bolt clamps the ends of the split boss and it engages in a groove round the centre of the eccentric. Rubber sealing washers are interposed between the eccentric and the screwed bushes. Adjustment of the camber angle is effected by slackening the centre pinch bolt, removing the grease nipple, and inserting a special spanner through the grease nipple hole into a socket cut for it in the end of the pin. When the adjustment has been made with the spanner, it is locked by tightening the pinch bolt.

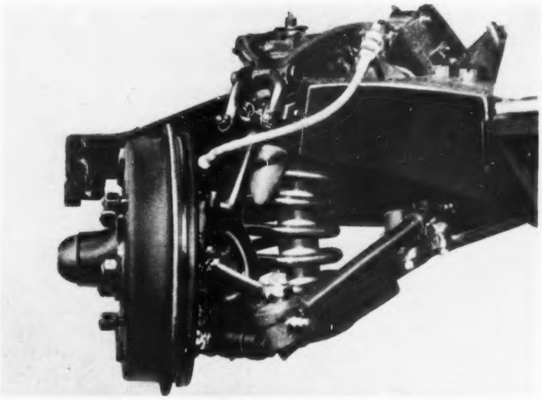
A two piece En 18T stamping is employed for the lower wishbone link. Each piece is of I-section and bosses are formed at each end. The two pieces are bridged by the lower spring pan and by the bump stop anvil. This anvil is a $\frac{1}{8}$ in thick pressing, the centre of which is dimpled to control the spread of the small section at the end of the bump stop rubber so that it is not



A Tufnol washer interposed between two steel ones takes the axial thrust at the swivel pin



ARRANGEMENT OF THE FRAME
All the main members are of 12 s.w.g. steel sheet



The front suspension unit viewed from the rear

unduly strained. It is secured by two $\frac{3}{8}$ in diameter bolts passed vertically one through each of the two parts of the wishbone. The lower spring pan is secured in a similar manner, by four $\frac{3}{8}$ in diameter bolts. Two studs are secured to the bottom of the pan to carry the bracket that supports the lower end of the shock absorber. This bracket is $\frac{1}{8}$ in thick and its cross-section is of inverted channel form, with the sides bent back through 180 deg and extended upwards to form the two lugs carrying the rubber bushed ring type end fitting of the shock absorber.

At the inner end of the wishbone the screw type bush and rubber seal arrangement is similar to that already described for the top wishbone. However, the pin is a plain bar of En 351 with its ends threaded. It is clamped in two split bosses, one immediately inboard of each bearing. Each boss is secured by a $\frac{7}{16}$ in diameter stud and a $\frac{7}{16}$ in diameter bolt to the flanges on the underside of the frame cross-member, one of which clamps the split boss on to the pin. The rubber sealing washers at the inner end of each bush bear directly against these bosses, so no retainer washer is needed. The bearing centres are $12\frac{1}{4}$ in apart and their effective length is $1\frac{5}{8}$ in.

At the outer end of the wishbone are the two lugs between which is the lower boss of the stub axle carrier forging. A screw type bush is pressed into this boss, and the $\frac{13}{16}$ in diameter Whitworth threaded En 32B pin is screwed through the rear portion of the wishbone and into the bush until a shoulder at the front end of the pin comes up against the front lug. In front of the shoulder, the pin is threaded $\frac{5}{8}$ in diameter and a split pinned nut on the end clamps it to the wishbone. At the rear of the pin, a lock nut is tightened up against the other part of the wishbone and a grease nipple is screwed into an axial hole in the pin. This hole extends as far as the centre of the bearing where a radial hole communicates with a groove around the pin. Rubber sealing rings are interposed between the stub axle carrier forging boss and the wishbone lugs.

The stub axle carrier forging is of En 18D. It incorporates two knuckles between which is carried a boss formed on the inner end of the stub axle. The assembly is held together by the $\frac{13}{16}$ in diameter En 18T swivel pin, which is passed through the knuckles and central boss. Midway between the two ends of this pin is a groove in which is engaged a cotter in the boss. This cotter furnishes the necessary axial location for the pin.

A thrust bearing is interposed between the upper knuckle and the boss on the stub axle. This bearing consists of a Tufnol washer sandwiched between two case hardened mild steel washers. All three are $\frac{1}{8}$ in thick by 2 in diameter. The two outer washers are dowelled, one to the knuckle and the other to the boss, while the centre washer is free to rotate. An inverted cup-shaped pressing completely enshrouds the bearing to screen it from foreign matter. A Hycar synthetic rubber ring is housed in a groove in the lower face of the boss to form the seal.

Both the STA/7/CPS swivel pin bushes are $1\frac{7}{16}$ in long, and each has a spiral groove cut in the bearing surface to distribute the lubricant which is supplied through a grease nipple in each knuckle. Their centres are $4\frac{1}{4}$ in apart and they are a press fit in the knuckles. The whole assembly is sealed by two steel plates, one at the top and the other at the bottom. Each plate is secured by two set bolts passed through it and screwed into tapped holes in the knuckle.

Two taper roller bearings, spaced $2\frac{1}{2}$ in apart, carry the wheel. They are mounted on the En 18D stub axle which is $1\frac{1}{4}$ in diameter where it carries the inner bearing and $\frac{7}{8}$ in where it carries the outer one. The bearings are assembled into the malleable cast iron hub, one from each end, and the outer races bear against shoulders in the hub. A nut and large diameter washer on the $\frac{3}{4}$ in diameter threaded end of the stub axle pull against the inner race of the outer bearing. They draw the whole assembly up against a distance washer between the inner race of the inner bearing and the stub axle flange. This washer is shaped so as to clear the $\frac{5}{16}$ in radius of the root fillet.

A Gaco E 90 oil seal housed in the hub bears on the periphery of this distance washer. The seal housing in the end of the hub is enshrouded by a grease collector ring secured to the brake back plate. A hole is drilled through the back plate to drain the lubricant away from the collector ring and clear of the brake drum. Four bolts secure the back plate to the stub axle flange. The brake drum and wheel are mounted in the conventional manner on a flange round the hub. A steel cap, over the wheel nut, is pressed into the outer end of the hub, to retain the grease and keep out abrasive matter.

The effective length of the $\frac{5}{8}$ in diameter En 45 anti-roll bar is $30\frac{1}{2}$ in. Its radius arm has an effective length of $5\frac{3}{4}$ in. The bar is carried in rubber bushes bolted underneath the brackets on extensions of the frame side-members forward of the front cross-member. The radius arms are bent to the rear and their ends are turned out parallel to the front of the lower wishbone link to which they are attached at a point about $6\frac{7}{8}$ in from the bearing centre. The attachment takes the form of a rubber bush, which has sufficient resilience to allow for axial differential movement between the two components, in a pressed steel cap secured to the wishbone by two bolts.

Steering

A Burman recirculating ball steering gear is employed. The ratio, which is variable, is 19·3 : 1 in the straight ahead position; and $4\frac{1}{2}$ turns of the $17\frac{1}{2}$ in diameter steering wheel are required to move it from lock to lock. The wheel angle on the inside lock is 31 deg, and on the outside lock 29 deg.

The steering box is mounted on the frame-side member about 8 in behind the wheel centres, and a three piece track rod system is employed. All rods are built up from 0·865 in outside diameter by 0·615 in inside diameter tube. The tube ends are split, and Alford and Alder self-adjusting spring loaded ball joints are screwed in and locked by bolted clamping collars. The track rod length from centre to centre is 18·92 in and the side rods are 14·1 in long. The sump is stepped to clear the track rod. Both the drop arm and the idler lever are of En 15S and they are symmetrically arranged on the frame. The ball joints for the track rod are attached to the end of the lever and the drop arm at a point $9\frac{1}{2}$ in from the pivot centres, and the joints for the side rods are attached 7·35 in from the pivots. The effective length of the En 18T steering arm is 8 in, and its inner end is tapered and keyed in the boss of the stub axle. It is secured by a $\frac{5}{8}$ in diameter slotted nut.

Brakes

Lockheed hydraulic brakes are fitted. At the front they are of the two leading shoe type, and at the rear, a leading and trailing shoe is employed. The drum diameter is 11 in, the shoe width $2\frac{1}{4}$ in, and the shoe lining area $95\frac{1}{2}$ in^2 at the front and the same at the rear. A lever type hand brake control is pivoted on the scuttle. Its effective length is approximately 10 in and a hole is punched in it about $1\frac{3}{4}$ in from the pivot point to take the bent over end of the tie rod. These dimensions give a lever ratio of 5·95 : 1. The other end of the rod, which is also bent over at 90 deg, is attached to a relay lever on the front of the dash. The motion

is transmitted by another rod to a lever carried on a bracket on the body. This lever is centrally pivoted, and pinned to its lower end is the cable control for the rear brakes.

The cable is taken through a conduit, one end of which is attached to the rear arm of the cruciform bracing and the other end is secured to the bracket carrying the compensator device on the back axle. This compensator is a pivoted lever and swinging link arrangement. The swinging link consists of two plates, the ends of which are carried on the ends of two ⅜ in diameter pivot pins bearing in Vandervell bushes pressed into tubular housings. One of these housings is welded at its centre to the bracket under the back axle, and the pivoted lever is similarly attached at the centre of the other housing. The bearings are sealed with felt rings in recesses formed by counterboring the ends of the housings.

A T-shaped pivoted lever is employed. The adjustable end on the cable control is pinned to the leg of the T, and the ends of the brake control rods are bent over at 90 deg and carried in the ends of the arms. The outer ends of the brake control rods are passed through a U-shackle, and screwed into a threaded saddle piece inside the U. A lock nut on the rod is tightened against the other side of the U to fix the setting. The shackles are pinned to the levers on the brakes. The overall lever ratio is 13·4 : 1.

The pivot pin for the brake pedal also carries the clutch pedal and is formed by a BS 980 C.D.S.—3 tube. It is secured by tension pins in a tube on the frame. The pedals are fitted with cold rolled En 2 perforated bushes in the ¼—½ hard condition. These bushes are 1¼ in long by ⅞ in inside diameter. A lever ratio of 5·88 : 1 is obtained with 6 $\frac{7}{16}$ in pedal travel. The brake master cylinder is bolted to a bracket welded on one of the front arms of the cruciform bracing.

Frame

The frame is of simple construction and it is built up almost entirely of 12 s.w.g. channel section members. Three cross-members only are employed. The ends of the front one are swept back and its centre is dropped so that it will clear the fan pulley, while still providing adequate support for the suspension units. It is of top hat section, 6 in wide by 4 in deep, closed by a plate welded to the bottom flanges, and a doubling plate is welded under the centre. Above it, on the front face, is the bracket supporting the cradle that carries the radiator. This cradle is of channel section and its ends are turned up and attached on each side to the body. The radiator is supported on a single rubber mounting in the centre of the cradle.

A channel section rear cross-member is employed. It is 2½ in wide by 3 in deep. The only other cross-member is an inverted channel section, 3 in wide by 2½ in deep, and it is positioned approximately 3 in forward of the rear axle to support the upper ends of the telescopic shock absorbers.

A large cruciform bracing is employed at the centre of the chassis. Each member is approximately 6 ft 10 in long and it is a fabricated I-section the overall dimensions of which are 2 in wide by 5 in deep. The thickness of the centre gusset plates above and below the cruciform is 12 s.w.g. At the centre, the width of the frame between the side-members is 39½ in.

Channel section side-members are employed. They are approximately 6 in deep by 2½ in wide at the centre. Towards the rear, where another channel section is welded in to form a box section behind the point of attachment of the cruciform bracing, they are swept inwards about 1¼ in, over the length of the spring, and then outwards again to where the rear bumpers and jacking supports are attached. At the front of the frame, they are swept in about 7 in from in front of the cruciform to a point in line with the wheel axes, and then outwards again to carry the bumper irons and jacking supports.

The channel sections are boxed in from the extreme front end to the point where the cruciform bracing is attached. The closure is effected at the front by another channel section welded inside the side-member. This channel is bent slightly towards the centre of the frame at its rear end and welded to the front cross-member. This enlarges the box section and forms a gusset to the front cross-member. Behind this, another channel section is welded in the side frame. The front end of this channel is bent towards the centre line of the frame to form a gusset to the cross-member. From a point about 12 in further back, where the frame side-member is swept out, the inner channel is continued straight to the rear and its end is welded to the front arm of the cruciform bracing. Where the two channels part company, a 12 s.w.g. gusset plate is welded on top and below. Closure of the frame side-member between the points where it is swept outwards and where it is joined by the cruciform bracing is effected by welding in another 12 s.w.g. channel.

Two brackets to carry the rear engine mountings are welded to the inner face of each of the two front cruciform arms. Another pair, for the steering box and idler lever, is welded to the inner face of the inner channels of the side-members just forward of the point where they part company with the outer channels and extend straight to the rear.

The front engine mountings are bolted to brackets welded in the corner between the side-member and the rear face at each end of the front cross-member. Seven body mounting points are incorporated on each side. The first and second as well as the fourth and fifth are 12 s.w.g. brackets overhung from the side-member. Between these is the third, which is a bolt attachment to the side-member. The sixth is a simple top hat section bracket welded on top of the frame side-member above the axle, and the seventh mounting point is on the rear cross-member.

Electrical equipment

Lucas electrical equipment is used throughout. The 12 volt, GTW 11A battery has a 64 amp-hr capacity at a 10 hour rate. It is charged by a C45 PV 5 dynamo used in conjunction with an RB106 voltage regulator. A Lucas M45G starter with a 9 tooth pinion is employed. The HV12 coil serves a DM 6 contact breaker and distributor, which has a centrifugal and vacuum controlled advance and retard mechanism. Champion NA8 sparking plugs are recommended. The head, tail, side and stop lamps are respectively 36/42W, 6W, 6W and 18W. Dual wind tone horns are used, and self-cancelling trafficators are mounted in the body centre pillar.

Other features

The spare wheel is stowed vertically on the right-hand side of the boot, and a Bevilift pillar type jack is provided. Jacking points are incorporated on the extreme ends of each chassis side frame. The exhaust system is built up in sections. A short section of pipe is bolted at one end to the manifold and is swept downwards and outwards to a point just in front of the cruciform bracing. Here it is flanged and bolted to another section of pipe which is carried straight through to the rear of the cruciform, swept round the rear shock absorber and over the axle and then straight back, alongside the tank, to the silencer. In order to facilitate the assembly of the pipe and avoid introducing strains into the system, the straight section between the arms of the cruciform bracing is broken, and joined by a 12 in length of flexible pipe. The silencer is mounted transversely immediately behind the rear cross-member from which it is suspended. Two Firestone vertical sandwich type mountings are used for the silencer, and a third one is bolted to a bracket on the side frame to support the pipe at a point just behind the flexible section in the centre.

'Dag' Dispersions

ACHESON COLLOIDS, LTD., 18, Pall Mall, London, S.W.1, have recently developed a number of dispersions of molybdenum disulphide. These, while retaining the advantages derived from a dry powder, are more mobile, more convenient to handle and capable of being readily blended with additional quantities of liquid carrier to facilitate their use under all conditions of industrial application.

To those interested in examining the properties of this new lubricating material, Acheson Colloids Ltd. are prepared to supply inspection samples of their "Dag" dispersions of molybdenum disulphide in mineral oil, alcohol, toluene, white spirit and several other carriers.

BRITISH PLASTICS EXHIBITION

Signs of a Growing Interest in Plastics Bodies for Motor Vehicles

THE British Plastics Exhibition is held once every two years, and whereas in the Exhibition held in 1951 there were no plastics motor bodies, this year two complete bodies were shown, a hard top and the front end of a third body, and a photograph of a fourth could be seen. Considerable publicity has been given in the United States to the application of plastics to motor bodies, and it would appear that interest in the subject is more widespread in this country than is generally believed.

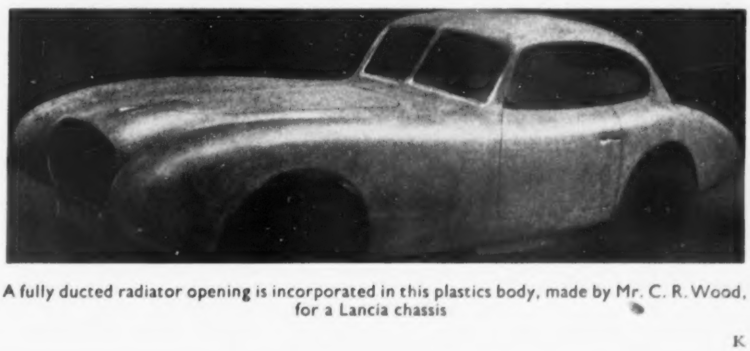
Weight for weight, plastics are generally more expensive than more conventional materials but they may still be used for two reasons. Firstly, they have a number of properties such as corrosion resistance, anti-drumming characteristics, light weight, resistance to denting, and ease of repair which make them attractive to the motor manufacturer. Secondly, for small quantity production, fabrication is less expensive because tooling costs are considerably less and labour costs are not quite so high as when panel beating is resorted to. It would even be rash to state dogmatically that the material is not suitable for mass production methods. There is evidence to show that production techniques are gradually being evolved which lend themselves well to mechanization and saving in time and labour. The principal obstacle at present seems to be that raw materials are not available in large quantities.

Perhaps one of the most promising developments from the point of view of saving in production time was demonstrated on the Ministry of Supply stand. It has been found that by passing an alternating current through the laminated plastics the curing time can be reduced to a matter of a few seconds. Another important result to be expected from the adoption of this method is that the expense of incorporating heating elements in the moulds would be avoided. The method is most easily applied to Durestos materials, but it is probably possible to apply it to glass fibre reinforced plastics by mixing with the resin a substance that will improve the electrical conductivity.

Mouldings made by the pre-form technique were shown on the stand of Ashdowns Ltd., of St. Helens, Lancs. This is a process that would lend itself well to mechanization and to the elimination of laying up by hand. In this process the glass fibres are used in the form of rovings, that is, they are chopped into lengths of about 1½ in. A perforated male mould is used, the perforations being closely spaced. Air is pumped away from the interior of the mould so that it is continually being drawn through the perforations, and as the rovings are applied they are held on to the mould by suction. In order to ensure that a layer of constant thickness is obtained, the mould is surrounded by a box into which the rovings are dropped and air jets are directed into the box to distribute them evenly. The rovings may be chopped *in situ* from reels of loosely spun yarn, or they may be introduced to the machine in the ready chopped condition. When a layer of appropriate thickness has been obtained, resin is sprayed on and partly cured. The shell is then removed from the perforated mould and placed between matched moulds with which pressure is applied until the cure is completed.

No information was available as to what material the moulds were made of, but a number of possibilities are apparent. For instance, it would appear that Kirksite might be suitable if production were to be on a large enough scale. Perhaps one of the most important fundamentals so far as moulds are concerned is that their surface finish is imparted to the plastics product. Therefore, if a product finish that needs no further treatment is required, the surface of the mould must be in a highly polished condition. Even more important is the nature of the surface after the parting agent has been applied.

On the stand of Scott Bader & Co. Ltd., there was a front end of a body for



A fully ducted radiator opening is incorporated in this plastics body, made by Mr. C. R. Wood, for a Lancia chassis

the Allard Monte Carlo saloon. It was moulded by Microcell Ltd., of London, W.C.2, using Marco resin. Another exhibit on this stand was a sample mould and a laminate prepared from it. Both the mould and the laminate were of glass reinforced plastics. The laminate was remarkable for its extremely smooth surface. In fact, the finish was of such a quality that no further treatment, such as polishing or painting, would have been necessary had it been part of one of the less expensive types of car. For a car in the luxury class, however, a coat of paint might have been desirable.

The finish is obtained by first laying a coat of heavily pigmented resin on the mould. Then, with this coat partly cured, the glass mat is laid on and resin impregnated and the cure completed. It is considered advisable, if paint is subsequently to be applied to the surface, to match the colours of the paint and the pigmented layer of resin so that if the paint should become scratched the damage would not be markedly noticeable. On the other hand, when applying paint to such a surface, it is difficult to see whether or not the coverage is complete, and there is a danger of patches being left bare.

Also exhibited on the Scott Bader stand was a photograph of a plastics body on a Buckler tubular frame chassis. It was made by the Galt-Glass Division of Durasteel Ltd., of Greenford. The body, which was laid up in the same manner as the samples already described, weighs only 74 lb. Glass cloth was used instead of mat because the resultant laminate is stronger, and therefore a thinner section may be used. Moreover, although the cloth is more expensive, it does not absorb so much resin as does the glass fibre mat. The boot lid and the doors were made from two pieces moulded together. It is claimed that an exceptionally good finish was obtained but in order to attain a very high standard, a coat of paint was applied.

Microcell Ltd. were showing a hard top for an Austin A40 sports model. It had been manufactured in conjunction with Vanden Plas England (1923) Ltd., who, in order that it might be shown in the Exhibition, had removed it from a car on which it had been installed for some considerable time. The finish of this hard top, which was made with Marco resin and glass cloth, is exceptionally good.

This body, supplied by R.G.S. Automobile Components Ltd., is fitted on an Atalanta chassis but, in order that its length may be adjusted to suit other chassis, it is made in two pieces together with a separate centre portion

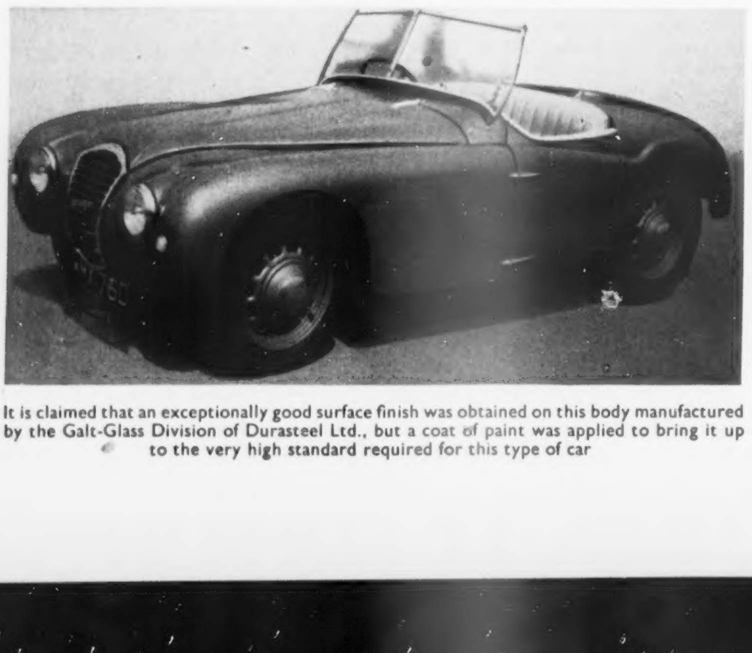
Another car body was displayed on the stand of Bakelite Ltd., London, S.W.1. It was made for R.G.S. Automobile Components Ltd., of Winkfield, Windsor, by the North East Yacht Building and Engineering Co. Ltd., of Blyth. Bakelite polyester resin and a glass fibre mat were used for its construction. The finish of this body was not so good as that of the others described, but it is supplied as part of a kit of components from which the purchaser can build up his own car and finish it as he wishes. A striking illustration of the resilience of this material was given by the demonstrator on the stand who kicked the body with considerable force and no damage resulted. The body is made in two pieces, a third piece being added later at the centre to adjust the length of the body to suit the chassis on which it is to be mounted. R.G.S. Automobile Components Ltd. recommend that flexible paint, manufactured by International Paints Ltd., of Co. Durham, be used on the body.

The other body which was shown at the Exhibition is for a Lancia chassis. Details of this body were given in the June, 1953 issue of *Automobile Engineer*. The outstanding feature of this exhibit is that it is a saloon body, whereas nearly all the other plastics bodies that have been produced anywhere in the world are sports models. Shown on the same stand was a 30 ft long glider wing moulded in one piece and a 14 ft diameter radar reflector.

British Industrial Plastics Ltd., of London, W.1, were displaying a plastics cab door for a truck made by a well known manufacturer. The door that it was designed to replace was a panel beaten one, and so the plastics version can, no doubt, be produced at a competitive price. Moulded in, square section tubes of plastics are used for the framework, and timber inserts are positioned at points where hinges, etc., are attached by means of wood screws. The panels are made of Fibreglass chopped fibre mat bonded with Laminac 4128 which the exhibitors manufacture under licence from the American Cyanamid Co. In order to reduce the cost of the door, 30 per cent of powdered chalk was used as an inert filler in the laminate.

Several interesting exhibits were displayed on the Shell Chemicals stand. Among these was a special spray gun, made by B.E.N. Patents Ltd., for applying resin. The resin and accelerator are in separate containers and are mixed in the nozzle of the gun which in other respects is similar to most conventional, compressed air units. No doubt the mixture obtained is more homogeneous than that produced by the usual method of mixing the resin and accelerator before applying them in a gun. Moreover, appreciable time saving is obtained by the elimination of the mixing operation.

Among other exhibits on this stand were glass cloths ready impregnated with Epikote resin for dry lay-up laminating. So far as is known, the storage life of this impregnated sheet is at least several months. The minimum curing temperature is 145 deg C. Good laminates can be made at contact pressures, although a marked improvement in strength is obtained at pressures of 25 lb/in^2 or higher. If excessive flow out of resin is experienced, it may be necessary to pre-heat at contact



It is claimed that an exceptionally good surface finish was obtained on this body manufactured by the Galt-Glass Division of Durasteel Ltd., but a coat of paint was applied to bring it up to the very high standard required for this type of car

pressure. This pre-cure time required will vary with the thickness of the laminate, the reinforcing material, curing temperature and final pressure. Normally the time required for curing laminates ⅛ in thick is approximately 30 min at about 165 deg C, and 1½ to 2 hours with a thickness of ½ in. With ⅛ in thick laminates, about 92 per cent of maximum strength is obtained in 15 min at 165 deg C, or in 30 min at 145 deg C.

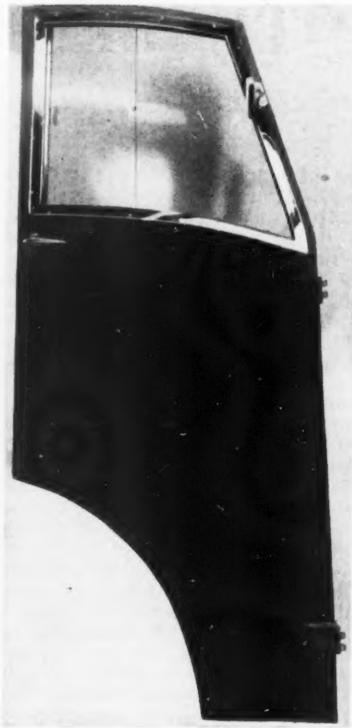
Another Shell Chemicals product that was shown was Epikote Adhesive VI. This can be used to bond together components made of any of a number of metals, such as aluminium, steel, brass, copper and magnesium. Either liquid phase or vapour phase degreasing processes may be used to clean the surfaces to be bonded. Surface preparation, such as etching, pickling, etc., is not essential though it may give better bond strengths than those obtained with mechanically prepared surfaces. A thin layer of adhesive is spread evenly on each of the surfaces to be bonded. These are then pressed gently together, only contact pressure being required. It may be cured at room temperature, that is, 25 deg C, and 20 per cent of the ultimate bond strength is developed in 24 hours. In most cases this partial cure will permit the removal of the clamps used to hold the assembly together. Handling strength is developed in about 2 days, and a full cure is reached in about six days. At 95 deg C a cure can be effected in 45 min. Moreover, curing may be completed even more rapidly at higher temperatures, but under these conditions the time factor is critical.

Bakelite Ltd. were showing Vybak transparent rigid sheet which is suitable for side screens on convertible cars. The flexible transparent material VB215 was also displayed by this manufacturer. It is used by some manufacturers for rear windows in hoods for convertible models. The manufacturers claim that it will maintain its flexibility and will not crack, age harden or discolour either when folded up with the hood in the stowed position or when extended.

Resins for Shell moulding were displayed by Bakelite Ltd., and James Ferguson and Sons Ltd. The Shell moulding process was described in the February, 1953, issue of *Automobile Engineer*. With the Bakelite process, fairly fine sand is recommended ; the bulk should pass through 60 mesh but nearly all should be retained by 200 mesh. Since only about 5 per cent of resin is added to the sand, thorough mechanical mixing is essential. It has been found that a blade or paddle type mixer is satisfactory for this purpose. The distribution of resin through the sand is materially assisted by mixing 0·05 per cent of wetting agent into the sand before adding the resin.

The general purpose formula recommended by James Ferguson and Sons Ltd., is as follows :

¼ to 1½ per cent starch
0 to ½ per cent clay



An inert filler was incorporated in this moulding to economize in resin

1½ to 2½ per cent water
0 to ¼ per cent release medium
1 to 3 per cent liquid resin.

These are mixed with sand to make up 100 per cent by weight. The dry strength of the mould is proportional to the resin content, whereas the green strength is determined by the starch and water content. With Stadex 608 starch, 2 per cent of water has proved satisfactory but the quantity required varies with the grade of starch employed. The addition of clay also improves the green strength and tends to prevent sagging, although the incorporation of too much should be avoided since it absorbs resin and reduces the dry strength. It may be introduced by including in the mixture a proportion of clay bearings and, such as Mansfield Red, or by using a Bentonite. The release medium may be commercial paraffin or any of the proprietary brands sold for this purpose.

Holoplast Ltd., of New Hythe, Kent, were showing Decorplast plastics panels. These are available in a number of extremely fine finishes and colours. Although they are expensive, no painting is required ; this is important not only from the point of view of production costs but also because there is no maintenance work such as painting or polishing to be done during service. These panels might be suitable for certain applications in luxury coaches.

Another exhibit on this stand was Holoplast panels. These consist of two panels spaced apart by moulded-in plastics webs. They may be constructed to almost any thickness that is likely to be required, and might be suitable for partitions, such as are used on some continental coaches to form a division between smoking and non-smoking compartments, or they might be suitable for sliding doors. Again, they are expensive, but in this application not only are maintenance costs eliminated but a minimum of fabrication is required to construct from them a flat door or partition. Moreover, they are fire resistant and have good sound and thermal insulating properties.

Welding machines for joining thermoplastic sheets were demonstrated on the stand of Radio Heaters Ltd., of Wokingham. The welding machines consists of two parts, a generator which converts the alternating current mains supply into high frequency power, and the mechanical system that clamps the material between the electrodes. The rise of temperature up to the fusion point of P.V.C. may take place in as little time as 1/10 second. Moreover, it is claimed that since the surface is gripped by relatively cold metal electrodes the highest temperature is obtained at the centre of the joint where the fusion is effected, while the surface is hardly effected at all. These machines may be used to replace sewing and other methods of making joints in a number of applications where plastics sheets are used.

The reversible ventilator unit shown by Vent Axia Ltd., London, S.W.1, was remarkable for its silence. Fan silence and efficiency is an essential feature in heating and ventilating systems. Both are, to a large extent, dependent on blade shape and the use of suitable guide vanes. It is possible that these shapes may be more easily reproduced in plastics than in metal. For applications to motor vehicle heating and ventilation systems where extreme silence is required, components of this type might be supplied as optional extra fitments.

Simmonds Aerocessories Ltd., of Glamorgan, were showing their range of Spire nuts and clips. One of these is most suitable for fixing plastics badges, etc., to metal panels. This is the Spire Speed Fixing for non-threaded components and it resembles the Spire Speed Nut for threaded screws. A circular section dowel or a rectangular section projection on the back of the plastics component is pushed through the fixing unit, and is firmly held by it. The unit consists of a small rectangular spring steel plate slotted transversely at its centre. It is also split longitudinally at each end of the transverse slot in such a manner that the slot and split resemble an H, and the metal remaining between the arms of the H forms two tongues. The dowel or projection of the plastics component is pushed through the slot and is gripped by the two tongues which are pushed forwards in such a way as to prevent it from coming out again. The ends of the tongues are suitably shaped to accommodate the projection.

DUNLOP TUBELESS TYRE

An Interesting New Development

THE introduction of another type of tyre to the Dunlop range was announced at Fort Dunlop on the 21st May. This is a tubeless tyre, the construction of which is similar to that of the standard product, except that an air-retaining rubber lining has been bonded inside and a strip of puncture sealing rubber is attached to the inner face of the lining below the tread. The air retaining lining is turned round the bead and terminates on the outer face of the tyre where it bears against the wheel rim flange.

This tyre cannot be used on wire braced wheels or where separate flanges are incorporated, because the seal is effected simply by the air pressure forcing the bead of the tyre against the flange. Therefore there must be no holes in the rim through which the air might escape. A special valve is employed. It has a flange, or head, on the inner end of its housing, and is pushed from the inside of the rim through a rubber grommet in a hole in the side of the well. A nut is then screwed from the outside on to the body and tightened against the rubber grommet to effect a seal. In all other respects the valve is similar to the standard product.

Three main advantages are claimed for this new development. The first is that because the inner lining is not under tension, it is nearly twice as effective as an inner tube in forming a barrier against loss of pressure. Secondly, the inner lining protects the carcase to a certain extent against damage due to running in an under-inflated condition. This protection is also effective in the event of accidentally hitting an obstacle such as a curb when the tyre is under-inflated. The third advantage arises from the incorporation of the puncture sealing layer. The material of this layer is of such a composition that it will cling to a nail or any other object which may pierce the tyre, and prevent the air from escaping. There is reason to suppose that the resistance of the new tyre to bursting as a result of damage to the carcase is greater than that of a normal one.

At present the new unit is being manufactured on a small scale, and the price is the same as that of a conventional tyre together with its inner tube, plus 20 per cent. It is probable that if production is undertaken on a large scale, the price of the new tyre will be the same, if not less than, that of the standard product. In actual fact, the manufacture of covers and tubes separately has been brought to such a height of efficiency that there is little scope for reduction in manufacturing costs. At present the tyre is being produced in sizes ranging from $5{\cdot}00 \times 14$ to $6{\cdot}50 \times 16$ and $5{\cdot}20 \times 13$ C.T. to $7{\cdot}60 \times 15$ C.T.

It is possible that nails which penetrate the tread will ultimately cause serious damage even though no loss of air pressure occurs. For this reason the manufacturers recommend that tyres should be inspected at intervals of every two or three thousand miles, and that any nails or other sharp objects which have been picked up should be removed. Small objects may be removed and no air leak will follow, but if a substantial hole has been made it is likely that pressure will be lost. Then, the procedure is to mark the position of the hole, take the tyre off the wheel, and plug and patch the hole from the inside.

Fitting the tyre calls for a certain amount of skill. Although the operation is probably easier than with a conventional tyre and inner tube, great care must be taken to ensure that the rim is smooth. This is not difficult with a new rim, but with an old one a wire brush or emery must be used to remove the flakes of paint, rust, etc. If the flange has been distorted locally, it must be hammered straight. In extreme cases, on burrs which may have been thrown up on the rim as well as on any high spot at the butt-welded joint, it may be necessary to use a file. Finally, the flange and bead seating should be wiped clean with moist rag.

Before fitting the tyre, the valve is inserted in the hole in the well and sealed with its rubber grommet. The valve core must be removed. Next the tyre beads and rim flanges are moistened with clean water and the tyre is fitted in the conventional way, using levers only. The main precaution to be taken is to ensure that the beads are not damaged. This means that levers should be in good condition and free from burrs, and unduly thick or wide ones should not be used. When fitting the final bead, the operation should be performed in such a way that the last part to go over the rim flange is that nearest the valve.

Then, in order to snap the beads home against the rim, the crown of the tread should be bounced at various points about its circumference on the road. Next, with the valve core still out, the air line should be connected so that the beads are forced home on their seats by the rush of air into the tyre. It should be bounced again on the crown with the air line still attached. After this, the air line should be removed, the valve core inserted and the tyre inflated in the normal manner.

When a foot pump is used for inflation, the valve core is placed in position as soon as the tyre has been fitted, and a rope tourniquet is tightened around the tread to force the beads against the flanges. The tyre is then pumped up in the normal way. Whether a foot pump or a pressure line is used for inflation, a test should be made for leaks around the flanges. This is done after allowing the tyres to stand for a few minutes so that free air trapped between the flanges and beads may escape. The whole wheel may be immersed in a tank of water, or alternatively it may be placed horizontally and water poured into the channel formed between the flange and the tyre wall.

Removal of the tyre from the wheel is effected in the same way as with a more conventional one, except that there is no tube to be taken off. However, extra care is necessary when breaking the tyre away from the rim flange and on lifting the bead over it. This care must be taken in order to ensure that the bead is in no way damaged. If the tyre has been removed for mending a puncture caused by a nail, a rotational motion should be imparted to the nail when withdrawing it. This helps to draw up the puncture seal composition into the hole. After cleaning the area around the hole with petrol or naphtha, the solution should be allowed to evaporate and the puncture seal should be worked into the hole, which should then be covered with an ordinary repair patch.

A convincing demonstration of the fact that the tyre will do all that is claimed for it was given at Fort Dunlop when the announcement of its introduction was made. First, a Humber Pullman saloon on which these tyres were fitted was driven for 10 laps round a tight circle at 15 m.p.h., giving a side load of about $0{\cdot}5$ to $0{\cdot}6g$. Under these conditions, the wheels were on the point of sliding and the maximum sideways distortion of the tyre that can be obtained was applied. The pressure in the tyres was measured before and after and a scarcely measurable increase in pressure due to a rise in temperature in the tyre was observed.

Next, the car was run over a large number of nails on the road. These nails had been mounted on squares of cardboard so that they would stand point uppermost. After the vehicle had stopped, about eight nails could be seen in each tyre and all had penetrated right through the casing. After doing a few more laps under the same conditions as before, but with the nails still in the tyres, the car was driven with a full load of passengers on a test course for about half an hour. Periodic tyre pressure checks were made and the final one showed that, because of the increase in temperature, the pressures had risen from 24 lb/in^2 all round to 28 lb/in^2 on the two front wheels and 29 and 30 lb/in^2 respectively on the nearside and offside rear wheels.

COACHWORK DESIGN

A Review of the Outstanding Drawings in the I.B.C.A.M. Contest

NOT for many years have the annual drawing competitions organized by the Institute of British Carriage and Automobile Manufacturers, contained such a variety of subjects as were included this year. These competitions, which are supported by the Worshipful Company of Coachmakers and the Society of Motor Manufacturers and Traders, provide a valuable contribution to the traditional craft of bodybuilding, and it is evident that the closest association which exists between the parent Institute and the more recently formed section devoted to body engineering has been responsible for the introduction of a competition for a saloon body of chassisless construction with an engine not exceeding 800 c.c.

Another section calls for an arrangement drawing of a trailer caravan, and it is believed that this is the first occasion on which this type of vehicle has been the subject in one of these annual competitions. It is gratifying to note that those responsible for the furtherance of the art and craft of coach draughtsmanship are fully aware of the need to foster interest in entirely modern forms of construction and also of the need to develop ideas in connection with living vans.

Competition No. 1

This competition was open to persons of British nationality without age limit. It called for a coloured drawing and also an outline drawing showing elevation and half plan of a two-door, four-light convertible body giving seating accommodation for four persons. The quarter lights were to be arranged to drop flush with the top of the body and when open, the head fitting had to be concealed.

A scale of ¾ in to the foot and a three-quarter front view were specified for the coloured drawing. The entrant could, if he desired, use mechanical means for finishing the drawing. For the outline drawing a scale of 1½ in to the foot was specified. This drawing had to provide essential information for checking practicability, particularly in regard to the working of the hood and quarter light mechanism. The competitor was at liberty to select the method of hood operation, which could be either hand or power operated.

In this section the first prize of £50 and the Company's medal was awarded to C. G. Neale, whose drawing showed an excellent appreciation of the type of workmanship that is expected in illustrations intended to interest the potential purchaser. The coloured drawing, which is carried out in light green, not only gives the three-quarter front view required, but also has a high viewpoint that permits the full seating arrangements to be shown, or at least as much of these arrangements as is necessary to give the general seating plan.

The outline drawing in addition to the side elevation, half-plan and half-front view, also includes a cut-away view of the body only, that gives a very good idea of the method of hood operation. The linkage is actuated by hydraulic electric fittings with double acting head cylinders mounted on bell crank linkage located immediately in front of the wheel arch. Operation of the lid covering the compartment into which the hood falls is by twin jacks. An interesting point is that the division which separates the hood compartment from the luggage boot behind is composed of flexible material, which can be pushed forward to allow suit cases to occupy the space normally taken by the folded hood when maximum luggage accommodation is desired. In this case it is, of course, necessary to have the head rest. The head material is permanently secured below the panel surface to allow for link movement, and the rear panel of the hood contains a full width laminated plate glass rear light.

In general, the lines of this design follow typical modern styling, without distinct front and rear wings, and with the side panels carried through at maximum width from front to rear. These panels are attractively swaged to maintain the general impression of prominent horizontal lines which are a feature of this design, and the upper swaging passes across the quarter panels to finish in a built-out housing that accommodates the rear lamps. The lower line of the swaging also passes across the quarter panel to merge into the rear bumper arrangement, which constitutes a projection across the base of the rear body panel.

The front seat is of sufficient width

This design by C. G. Neale was awarded first prize in Competition No. 1

(59 in overall) to accommodate three passengers, and to give comfortable seating there are folding arm rests at each side of the central passenger. To give the fourth seat the competition calls for, there is a corner seat in the rear portion of the body, which has a removable cushion, enabling the seat to be used on either side. This cushion can be removed entirely, giving yet more space for the accommodation of luggage. For the occupant of this corner seat there is, of course, ample leg room, since he sits across the body; this arrangement also gives easy access to a sliding door situated below the hood compartment and disclosing when open, a storage space for picnic equipment.

Other features of this body include petrol fillers concealed in front of the tail lamps, there being a lift-up lid for this purpose, a cabinet and table in the back of the fixed centre section of the front seat, the side seats being made to tip, and a crash pad arranged across the top of the instrument panel.

The second prize was awarded jointly to two entrants, each of whom received £20 and the Company's medal. One of these competitors was R. D. Haynes, whose design was of more conventional style than that of the first prize winner, but was nevertheless, worthy of close examination. This is a design for all-metal integral construction and special attention has been paid to what this competitor evidently considers to be the many structural shortcomings of the convertible body. Of these shortcomings, the door design in particular has been the subject of special attention, and to overcome the tendency of the half-door to drop, alloy castings are used for the door shell, pillars and sills to receive and transfer the stresses to the body shell. By this means much of the weight normally associated with the strengthening of the under body is eliminated.

The design provides a door which when closed, bridges the gap of the door aperture and so produces an unbroken structure. A desirable degree of control flexibility is obtained by means of a hydro-electric system in which pressure created by an electrically driven pump, forces hydraulic fluid into rubber pads on the hinge and shut sides. These pads make close contact with the standing pillars. The whole arrangement is automatically operated by a micro switch that is integral with the upper door hinge. The hydraulic fluid is also forced into horizontally opposed cylinders arranged just below the waist line to provide a bolt action, not only on the shut side but also on the hinge side. The pressure is released both on the horizontal bolts and the vertical rubber pads by operation of a single control.

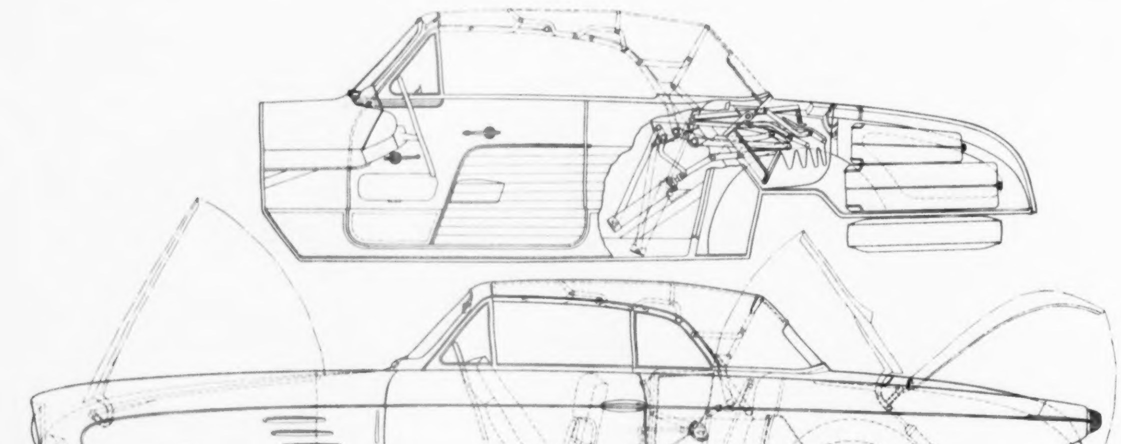
Although this competitor makes no mention of another major problem of convertible body construction, namely that of scuttle shake, the layout of the facia arrangement makes it obvious that good transverse bracing is possible. Moreover, since a broad shelf behind the instrument panel itself is flanked by triangular panels merging into the normal contour of the panelling at the rear end of the scuttle, there is little question that if adequate attention is devoted to the question of material stiffness, a very strong transverse unit could be formed in this area.

This competitor does not show a power-operated hood. Rather, he concentrates on a simple spring-loaded mechanism, but indicates the points at which power mechanism could best be introduced. In his layout for spring-loaded equipment, it is interesting to note that the operation of lowering the hood causes a link to tip the squab forward, at the same time supporting the passengers, thus allowing the rear light to fall within the hood recess. The hood is made in two sections and the design is such that it is possible to lower the rear portion of the roof into its recess without furling the material above the front seats; this gives the effect of the traditional landaulette head arrangement.

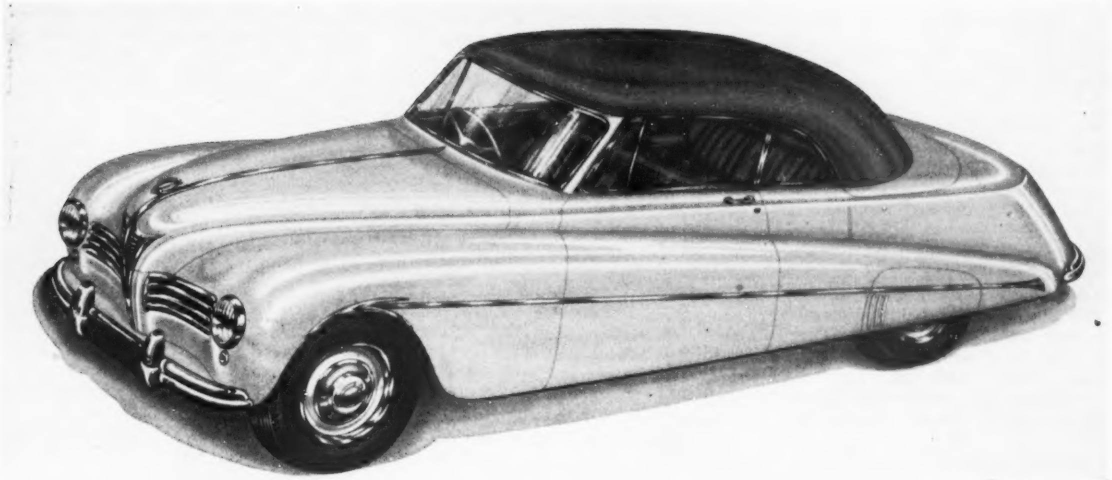
This design is notable for its very high front wings, which house the side and head lamps. The wings run through to the rear, at the full width of the body throughout. The seating arrangement follows conventional layout, but the front seat is divided centrally and the driver's portion has a neck rest which is combined with a tilting squab to give good support to the back. The other half of the seat squab can be folded forward and can then be used as a table, whilst the back seat squabs may also be folded down to enable luggage to be stored on the rear seat as well as in the normal luggage boot. When the rear seating compartment is used for the accommodation of luggage, it may be covered by a panel which can be folded and stowed under the luggage boot lid.

This competitor gives a fuel tank arrangement at the forward end of the boot above the rear axle, and also includes five-gallon tanks at each side of the propeller shaft and under the rear seat to give a reserve supply. These tanks can be rendered inoperative by taps in each of the feed lines. The coloured drawing entered by this competitor is taken from a very low viewpoint and gives very little idea of the general proportions of the car.

The other joint prize winner was R. A. Unwin, whose arrangement



Part of the layout drawing for the winning entry in Competition No. 1



A design by R. A. Unwin, joint second prize winner in Competition No. 1

drawing gives rather more constructional information, particularly with regard to hood and window operation. His design is perhaps not quite so attractive as those of the other two prize winners in this section, but it is nevertheless essentially practical and is characterized by the line of the front wing, which is carried across the door and quarter panels to terminate at a point just in front of the rear bumper which embraces the rear corner of the body.

The coloured drawing is carried out in yellow and green, the darker colour constituting the hood material. A chromium plated ornamentation runs for the entire length of the wing, beginning with an arrow-head above the front wheel and finishing in a flared end behind the rear wheel. The folding head is fully concealed when lowered and is operated by hydraulic rams acting on folding pillars. The main head centres are fitted with self-aligning ball bearings, and all other centres are bushed with phosphor bronze oil-impregnated bearings.

For the cant rails and main pillars aluminium alloy castings are specified and the link work is to be in steel. The hydraulic rams are situated between the wheel arch and the quarter pillar. It is interesting to note that hydraulic operation is also used for the door and quarter windows. The cover which conceals the hood recess is hinged on its rearward edge and is actuated by a central hydraulic ram, whilst the quarter head recess covers are opened by similar rams but are closed by spring loaded hinges. The hydraulic pump, motor and relay box which supply fluid under pressure to these various rams are mounted on the dash bulkhead.

Competition No. 2

This competition was open to persons of British nationality, without age limit, and called for a general arrangement drawing to a scale of 1½ in to the foot of a four-door, four-light saloon of chassisless construction, with an engine capacity not exceeding 800 c.c., and the design was to be suitable for quantity production. It was specified that the drawing must show the dimensions and position of engine, gearbox, and other chassis components, the locations of which were left entirely to the discretion of the entrant. Special attention was to be given to the suitability of the design for large scale production, with particular emphasis on economy in tooling and assembly costs. Sufficient information was to be given to indicate the various pressings used and the types of component and panel joints. In addition, full size sections were required for various parts of the body such as the screen, door pillar, and other main constructional items.

The first prize of £75 and the Company's medal was awarded to L. L. Reeves. This entrant's draughtsmanship is of a high standard and his outline drawing is supported by a number of other sheets giving considerable detail concerning the structural features of the design. Regarding the layout of the body, it is to be noted that the seating arrangement is well within the wheel base, and in fact the rear seat squab does not extend behind the forward edge of the rear door, thus the wheel arches do not encroach on the rear doors and the full depth of the cushion is disclosed when the rear door is open. To allow this seating arrangement, the power unit is forward of the front axle, the unit in effect being reversed with its radiator adjacent to the bulkhead.

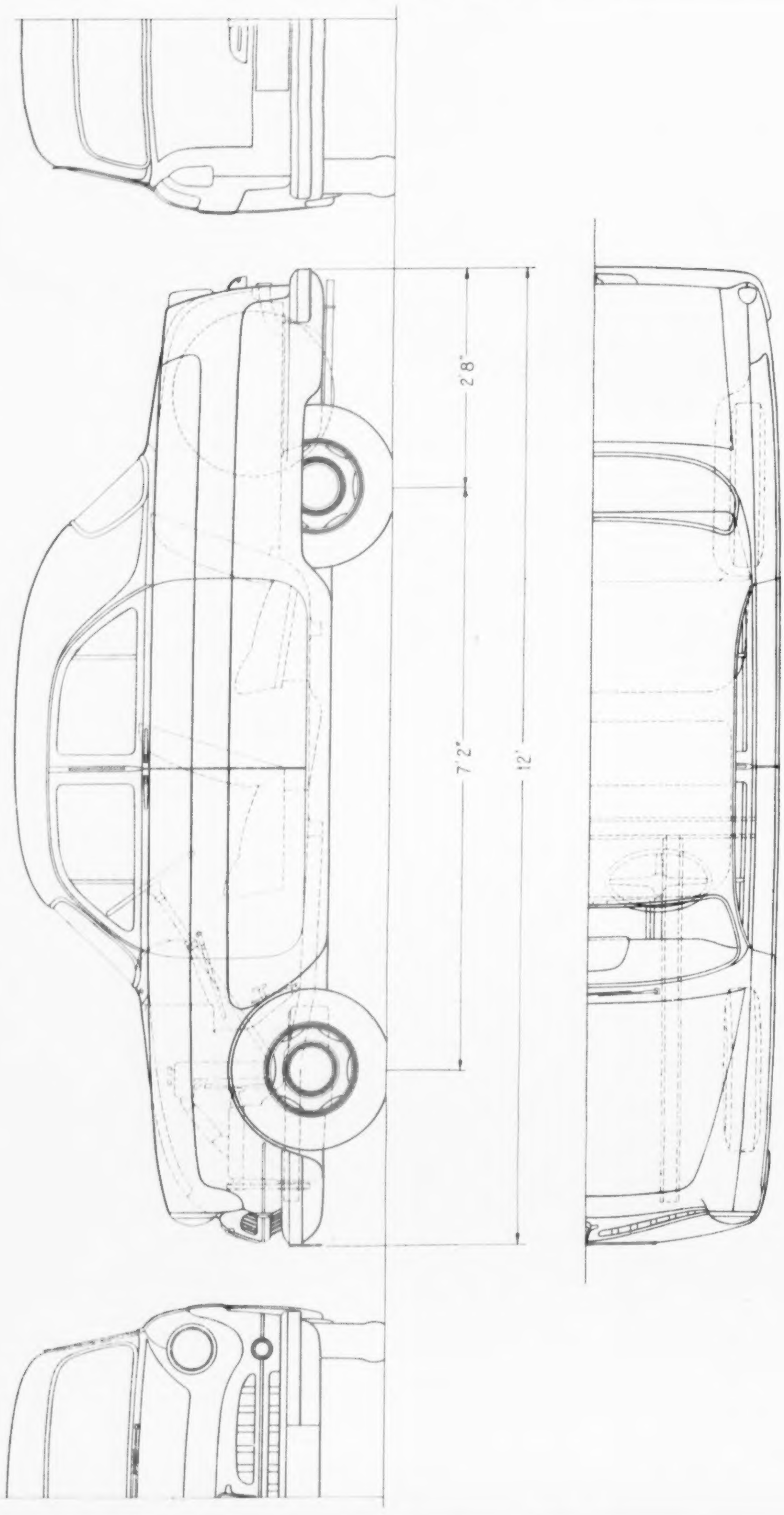
Several reasons are given for this particular layout, and it is claimed that with the engine in this position it is more easily cooled and there is less likely to be disturbance within the body from engine noise. Front wheel drive has been chosen on the claim that this would be cheaper to produce, with a saving in material and weight, whilst it also permits a flat floor within the body. The reason for mounting the radiator behind the engine is to give better weight distribution and to allow the bonnet line to sweep down lower towards the front, thus giving improved forward vision.

It is claimed that placing the seats well forward of the rear wheels allows a simpler construction, with economy in tooling and assembly costs. In the construction of the doors the four outer panels are made from two dies and another two dies are used for the four inner panels. The complete door assemblies are two right-hand and two left-hand doors, the right-hand front door becoming the left-hand rear door, and vice versa.

This arrangement obviously simplifies the question of glasses and reinforcements, and this particular door scheme is considered to be the main item of this entrant's design. Since the right-hand front door becomes the left-hand rear door, it is necessary to hinge the rear doors on their rearward edges, but this has the advantage of giving a better entry to the rear seats.

In his material specification this entrant calls for the outer surface panels in 21 B.G. mild pressing steel, and the construction units such as pillars, roof side rails, floor panel and under-body frame in 20 B.G. mild pressing steel. The reinforcements and spacers for under-body box members are in 18 B.G. and 16 B.G., whilst body fittings such as door handles and push-buttons are lightened wherever possible and consist of zinc base die castings. Glasses having single curvature are used for the windscreen and back light, and Triplex toughened glass is specified.

To facilitate repair work, the front and rear wings are detachable, and in the wing and door panels the moulding lines are formed to facilitate a two-tone colour scheme. The spare wheel is located on the left-hand side of the boot and is stowed in a vertical position which gives the best arrangement for luggage with access to the wheel when the luggage boot is loaded. To accom-



First prize in Competition No. 2 was given to this attractive design by L. L. Reeves

modate the spare wheel, a recess is formed in the floor and since the wheel is already in a vertical position, it can readily be rolled over the back rail which stands some inches above the boot floor. Since the rear seat is situated well forward of the rear wheel, there is ample accommodation for luggage within the boot, and in accordance with current practice the boot lid is of the lift-up type.

No second prize was awarded in this section, but an entry by D. V. Williams was given the third prize of £15 and the Company's medal. This competitor's design follows more conventional lines, and to give an appearance of extra length, the side panel surfaces are unbroken and run from head lamp to tail lamp without interruption. Maximum length of door lights is given by sweeping the rear door shut line back at a distinct angle, and the length of the front and rear door glasses is approximately the same. The design of the roof at its rearward end permits a square corner to the interior trim, giving extra head room. A wide back light with a flat glass is used, a flat glass being also used for the windscreen.

A four-cylinder horizontally opposed engine is used, since this competitor considers this to be the most compact arrangement, and with the gearbox well forward, maximum foot room between the front wheel arches is available. The front panel is detachable for servicing, and exends well forward to give protection to the engine. For added strength the front wings are welded in position. A diagrammatic view of the panel arrangement is shown, and mention is made of the fact that lead loading is used at the rear of the drip moulding only.

In this design the spare wheel is mounted in the boot behind the petrol tank, which in turn is behind the rear seat squab board. Thus, it would be more difficult to remove the spare wheel from the fully loaded boot, and it would, in fact, be necessary to remove the luggage before this could be done. Since a rear wheel drive is used, the floor is divided by a central tunnel.

The drawing submitted by V. W. Pondor was commended for which the award was £7 10s. Whilst the design is interesting it does not contain the structural information that is to be found in the other two prize winning drawings in this section.

In this design, the seats are again well within the wheelbase and the power unit is in front of the front wheel centre, with twin radiators in the front panel to permit an extreme forward mounting of the engine. The left-hand radiator is fan cooled and feeds the rear of the cylinder block, whilst the right-hand radiator has no fan and cools the front of the block. A central propeller shaft carries the drive to the rear axle, which is a composite unit embodying the four-speed gearbox.

The body has a well balanced appearance, since the front end panels, which embrace the engine and radiators, are approximately of the same dimensions as the rear panels. Both windscreen and rear light carry curved glasses and the upper panel above the engine is hinged at its rearward end. A point in connection with the combined differential and gearbox unit is that the gear control lever is in the centre of the floor, and hydraulic equipment is used to select the gears. Although the boot panels are of good length, the depth of the actual luggage compartment is reduced by the mounting of the spare wheel under the floor, where it must be high to give clearance for the rise of the rear axle unit. This spare wheel is mounted on a hinged platform that can be lowered when it is desired to withdraw the spare wheel.

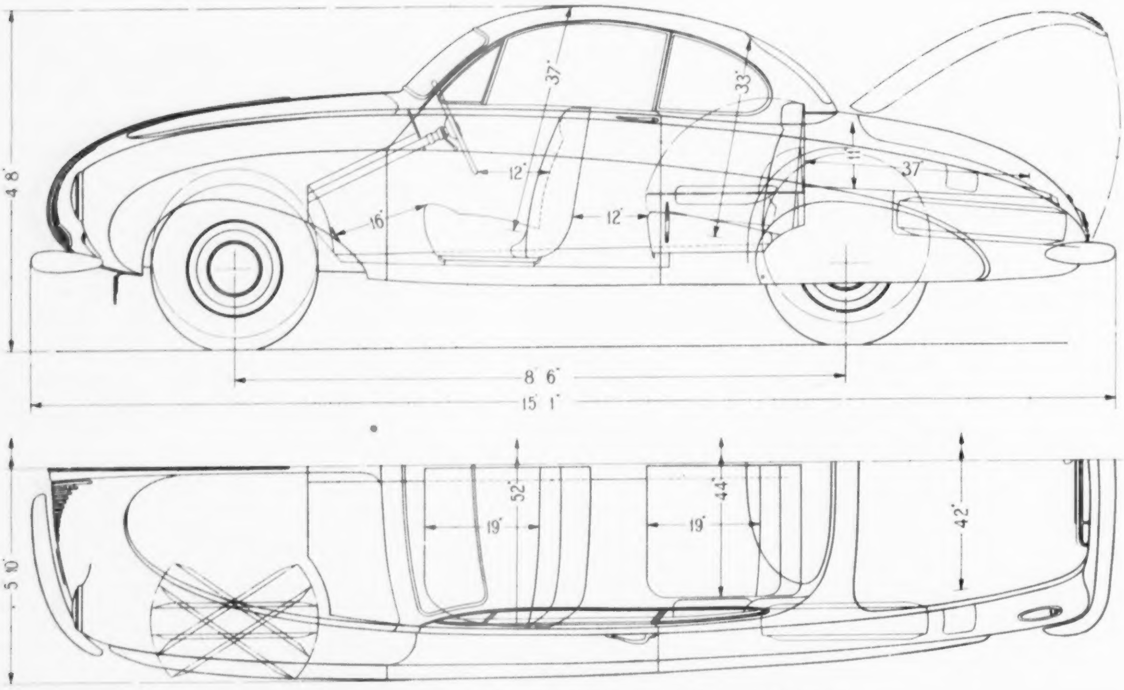
Competition No. 3

This competition was open to students of any technical school, or employees in body building shops in this country, under the age of 21. It was for an outline drawing of a two-door occasional four-seater sports saloon on a chassis with a wheelbase of 8 ft 6 in. The drawing had to be to a scale of 1½ in to the foot, in ink or pencil, and was to show the side elevation, half front and half rear views, and a half plan. Apart from the specified wheelbase, competitors could arrange their chassis details as they wished.

Some excellent work was seen in this students' section and the first prize of £20 was awarded to D. E. H. Chellingworth, who submitted a design having most attractive lines. Particular features are the well rounded and gently flowing bonnet panels, a treatment which is also carried out at the rear end for the boot compartment panels.

Although the chassis has a moderate wheelbase, the overall length of the car is 15 ft 1 in, this dimension being taken over the bumpers. This overall length makes possible the use of long well swept lines for bonnet and tail panels and for the wings, with the front wings extending well forward of the front wheels. The effect is good and the easy flowing lines of the roof structure harmonize extremely well with the scheme adopted for the lower panels. Both windscreen and back light use glass having double curvature, the windscreen of course, being a fixed component. The doors are hinged on the front pillars and a press

The winning design by D. E. W. Chellingworth for Competition No. 3



Side elevation and half-plan for the winning entry in Competition No. 3

button release for the locks is incorporated in the handles. In the sweep to the rear, the front wings are maintained at maximum width and there is no obvious break between the two wings.

A divided bench type front seat, each half being adjustable, comprises the normal seating accommodation. There are also rear seats for occasional use and a very neat inset drawing shows that each rear seat squab is capable of being hinged down to form a platform for extra luggage accommodation. To give protection, the back of each seat squab has metal runners, and the same treatment is adopted for the fixed squab board that divides the seat compartment from the luggage boot and prevents the entry of dust and fumes. The doors are equipped with drop glasses and the forward section of each window consists of a pivoting ventilation panel. Additional ventilation is obtained by hinging the quarter lights, whilst fresh air intakes to feed the heater and cooling system are incorporated in the front end styling.

The second prize of £15 was granted to N. Morris, for a design of rather more compact dimensions, the overall length being 13 ft 8½ in. Nevertheless, the design shows a pleasing line for the boot panels and rear wing, and the front end although finishing only a short distance in front of the wheel is quite attractive with its well rounded wing nose and harmonizing bonnet line. As a piece of draughtsmanship, the drawing submitted does not compare favourably with that of the first prize winner, but with its heavy outline it is quite a neat arrangement.

In addition to the outline drawings, a dimensioned seating arrangement is included, and this indicates that the knee room for the rear seat cushions ranges between 6 in and 10 in, according to the position of the front seat. Although this design is described as that of an occasional four-seater, it is felt that since there is so much room available in the luggage boot, it would perhaps have been better to have located the rear seat further to the rear and gained an inch or so more between the front and back seats. It is realized, of course, that in moving the rear seat as suggested it would no doubt have been necessary to raise the cushion line to clear chassis components, and this in turn would have increased the overall height of the body at this point, but nevertheless a compromise could have been reached that would have given a more favourable arrangement. To get the luggage capacity of 20 cu ft., petrol tanks are situated in the rearward portions of the rear wings, and the spare wheel is centrally mounted beneath the luggage floor.

A third prize of £10 was awarded to D. C. Coward, who submitted a drawing which quite definitely lacked the degree of draughtsmanship which is usually associated with prize winning drawings in these competitions, but the entry is particularly interesting because the designer has paid particular attention to the question of wind resistance. The two-door body is styled for minimum wind drag and maximum stability and road holding, and whilst the external appearance lacks the attraction of the more orthodox style of modern body, there are several features that hold the attention.

The frontal area has been kept to a minimum and all large panel surfaces are inclined to enable the wind ram pressures to be kept low. In the explanatory notes, the competitor states that the rear wing has a fin formation to create an aerofoil stabilizer to ensure lateral stability, but whilst no mention is made of maximum speeds, it is felt that a stabilizer of considerably greater dimensions is necessary to be of any real value. This competitor has obviously studied some of the many problems which arise when wind resistance and its effect on a road vehicle are being considered, and a light metal undershield is specified, an arrangement which is essential if full advantage is to be taken of the reduced drag on the upper portion of the body.

Magnesium alloy material is specified for the body framework and the main body panels. The roof panels are of moulded glass fibre and would therefore need little in the way of framing. Both the screen and back light are curved and are carried on to the sides of the body, a tinted glass roof panel being fitted above the front seats. The doors are exceptionally wide and although this competition did not call for any constructional information, there would obviously be difficulty in obtaining satisfactory hinging. The interior arrangements consist of a wide and divided bench type front seat and one seat against the off-side quarter panel giving a transverse seating position. Thus, with three passengers on the front seat, a total of four can be accommodated.

Competition No. 4

This competition was open to persons of British nationality, without age limit, and was for a full fronted double

deck luxury coach suitable for the dual role of attracting the tourist trade and for long distance travel. Overall dimensions to be 27 ft long and 8 ft wide, and the drawing was required to a scale of ½ in to the foot, showing the usual views and giving details of construction to a scale not exceeding 1½ in to the foot. Particular attention had to be devoted to the question of luggage accommodation and maximum observation from the interior of the vehicle.

The first prize of £75 and the Company's medal was awarded to D. G. Thaxter, whose 42-seat design was made to suit a rear engined chassis. This entrant's work is of a very high order and in addition to the usual general arrangement and detail drawings, a further sheet is included illustrating a partly sectioned view of the complete coach, very fully detailed and containing many explanatory notes. This illustration gives a three-quarter front view and is an excellent piece of work.

The external lines are attractive, since the windows are long and the cab panels are agreeably curved in plan. Large windows are introduced at each side of the cab. Further vision for the upper deck passengers is afforded by four large glass lights in the roof quarters, and the forward portion of the roof is also equipped with roof lights. All these roof windows are fitted with blinds.

Interior seating arrangements are interesting. Each seat is a separate unit, and the units are used either singly or in pairs. On the near side of the lower-deck there is first a pair of seats, followed by two single seats mounted at the side of the gangway to allow a luggage compartment to be arranged against the near side of the body. This is followed by two pairs of seats behind which is the flush fitting coach type entrance door. Behind this is a single seat for a stewardess, against the near side interior panel, and the rear side corner of the body is occupied by a roomy toilet compartment. The offside rear corner of the lower deck contains the stairway leading to the upper deck. Luggage, accessible from the outside, is stowed beneath the stairs. The space above the off-side wheel arch is taken up by a water boiler and a storage cabinet for packed meals.

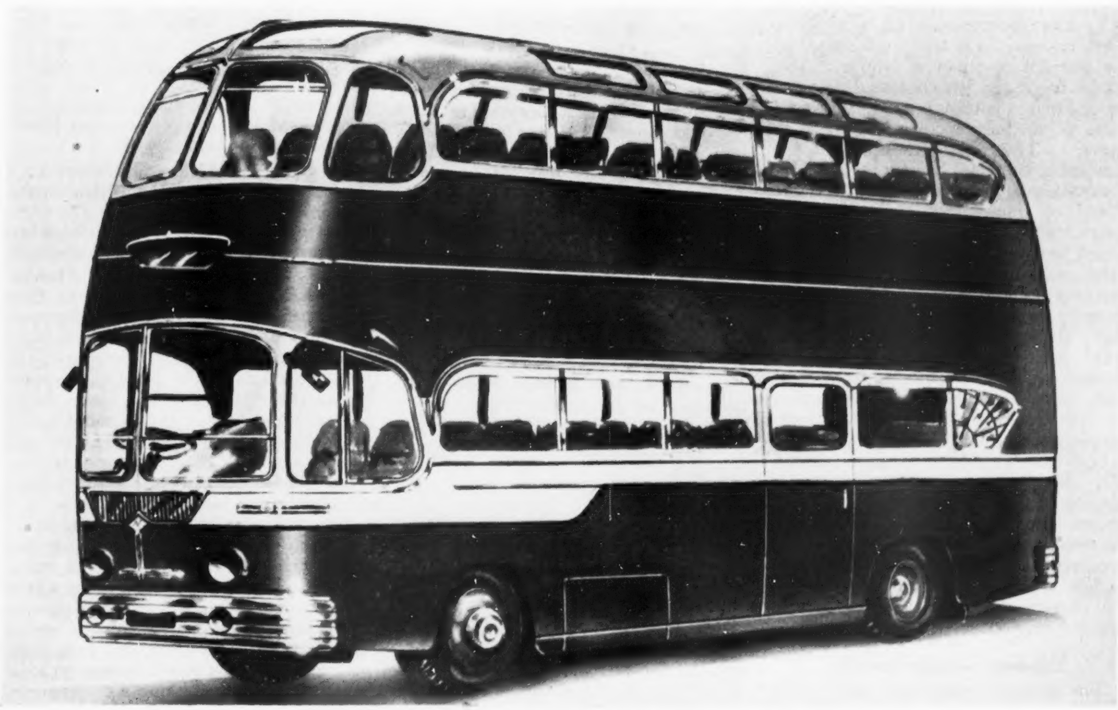
In the seating arrangement on the off-side there is a single seat behind the driver. This seat is placed against the gangway to allow for a storage space for luggage above the front wheel arch. There are then three pairs of seats before the emergency door is reached, opposite the main entrance door. The upper deck accommodates 27 passengers. Apart from one single seat towards the rear end, all upper deck seats are arranged in pairs. The stairs enter the top deck at the off-side rear corner. In addition to the luggage space already referred to, there are lockers below the floor on each side.

Metal framework is adopted for the construction of the body. It is packed with timber where necessary to facilitate the attachment of the panels and trimming. The lower saloon flooring is of ¾ in tongued and grooved boards, and there are traps to give access to chassis components, the entire floor being rubber covered. There is an intermediate floor of ½ in resin-bonded plywood, which is also rubber covered and rests on timber-packed steel bearers. The driver's compartment is separated from the rest of the vehicle by a light glazed partition behind the driver's seat which has a guard rail on the near side. An entrance door is provided in the off-side panel.

A comprehensive scheme of admitting either cool or warm air to the two main compartments is fully detailed, and a system is arranged whereby fresh air can be controlled to suit the individual needs of the passengers. An alternative scheme is shown, but this is less elaborate and does not permit the direction of air flow to be controlled with the same precision.

Notes are included giving the protective treatment that is recommended for metal parts, which are to be thoroughly degreased before bonderizing bath treatment. Metal surfaces that are in direct contact are to be coated with a Di-electric paint, and below the waist rail the inner surfaces of the panels are coated with a black bitumastic compound. All structural timber parts are dipped in wood preservative. For the flooring and chassis components, aluminium paint is recommended.

The second prize of £37 10s. and the Company's medal was won by J. Fyfe. This entrant's design has a central entrance door on the near-side, which gives access to a central gangway running fore and aft, leading to four double seats on the near-side, with a



D. G. Thaxter's winning design for Competition No. 4

courier single seat at the front, while there are five double seats against the off-side panel. Each seat has a low powered speaker built into the head rest, and the public address microphone is adjacent to the courier's seat.

This coach also has a toilet compartment in the rear near-side corner, the opposite corner being occupied by the staircase leading to the upper saloon. Beneath the staircase a roomy luggage locker extends to the chassis drop frame, and side luggage lockers are also incorporated. Two heaters are mounted in the front of the lower saloon, at either side of a demister unit, and the hot air from these heaters is ducted along the coves in both saloons, suitable outlets being provided with grilles adjacent to the seats. The warm air is transferred to the upper saloon by means of ducts mounted on the first pillars at both sides.

The third prize of £15 and the Company's medal was awarded to W. H. Timms, who entered a drawing showing a 44-seater body excluding accommodation for a spare driver and hostess. In many respects this design was very similar to that of the other prize winning drawings, one difference being that the toilet compartment is in the off-side rear corner, with a buffet and full depth luggage locker in the near-side corner.

The staircase to the upper saloon is on the off-side, almost opposite the entrance door, which is immediately in front of the near-side wheel arch. Double seats are used throughout and the question of extra vision for the occupants has received particular attention, additional shallow windows being used above the main windows on both upper and lower saloons.

Another respect in which this design differs from the other two is that behind the driver there is a waist high partition that, apart from a space of 15 in at the off-side, runs the full width of the vehicle, and since this chassis has an engine in the front, the space to the left of the driver cannot be occupied by seats.

Competition No. 5

This competition was open to persons of British nationality, without age limit, and was for a general arrangement drawing, in ink, of a trailer caravan completely equipped for not less than four persons. A special feature was to be made of the method adopted for exporting such vehicles, and the maximum overall length and width were to be 22 ft long by 7 ft 6 in wide.

Only a third prize was awarded in this class, and this was taken by C. R. D. Owtram, who receives £15. For shipping purposes the design can be broken down to six distinct units, one of which comprises the chassis with its timber floor and covered underside. The maximum dimensions are employed and accommodation is provided for five persons. One double bed is arranged transversely at the rear end. A third person can be accommodated on a single bed, which is formed on the off-side by using the back rest of the dining seat, which for this purpose is fitted with tubular legs. There is a further double bed on the off-side. The front off-side corner of the vehicle is occupied by a bathroom.

The kitchen equipment is in the near side front corner and the near side interior panel is occupied by wardrobes and other essential fittings. In the construction of the body, light alloy framing is used, although an alternative section for timber is given. The panelling is in 18 s.w.g. aluminium, and for insulation purposes fibre glass is used in the cavity between the inner and outer panels.

EXHAUST VALVES

Notes on Materials

EXHAUST valves have received more attention than any other component in the development of the internal combustion engine. The reason for this is the continual demand for increased output with improved thermal efficiency from a given size of engine. This may be achieved by increasing the compression ratio or by supercharging, both of which have the effect of increasing the maximum temperature reached during the cycle, which has two important effects on the valve problem. Firstly, it increases the valve operating temperature. Secondly, in order to avoid detonation or "pinking", an anti-detonant, usually tetra-ethyl lead, has to be added. The products of fuel, containing tetra-ethyl lead, have a virulent corrosive effect on the valve at high-temperature, increasing with the lead content.

Important problems in engine design are, therefore, to provide adequate cooling of the valve and valve guide to ensure correct lubrication, to choose the best materials to meet the operating conditions at economical cost, and to design a combustion chamber which will give good specific output with high thermal efficiency.

Exhaust valve materials

The desirable properties that should be aimed at in the development of exhaust valve material are :—

1. Resistance to scaling at valve operating temperatures.
2. Resistance to the corrosive attack at high temperatures of the products of fuels containing high additions of T.E.L.
3. Resistance to cold corrosion by moisture and by condensates containing bromides, etc.
4. High strength and hardness at valve operating temperatures.
5. Good-wearing characteristics.
6. Good heat conductivity to allow maximum heat transfer from the head of the valve.
7. Low coefficient of thermal expansion.
8. Ability to retain its original properties after frequent heatings to high temperatures.
9. Good fabrication properties, so that it may be readily forged and machined.
10. It should be easily heat-treatable to relieve strain and should be capable of being hardened at the tappet end to a high-hardness value.
11. It should be economical in cost.

Mechanical properties of some high-nickel alloys

Many of the high-nickel alloys have properties which make them quite suitable for this application. Outstanding amongst these is the high hot strength of the Nimonic Series of alloys, most of which are considerably stronger at valve operating temperatures than the steels commonly used. Between 700 and 800°C the hardness figures for the Nimonic alloys are in excess of the steels used for valve stems. The most important property of these high-nickel alloys is their outstanding resistance to corrosion, both in the cold condition and at high temperatures. This is combined with high resistance to scaling and high impact values.

Apart from the ordinary mechanical properties of the material, it is essential that it should be readily forged, easily machined and should require very little heat-treatment. This factor eliminates several materials which, though having suitable properties, present such difficulties in the manufacturing process that, unlike the Nimonic alloys, they cannot be considered an economic proposition.

SUSPENSION CONTROL

Some Aspects of the Application of Dynamic Absorbers to Suspension Systems

It would be rash to suggest that the development of the hydraulic damper has now reached the stage where further improvement is impossible. There are, however, strong arguments for taking stock of the present day operating conditions and analyzing anew the design problems to see if progress is along the right lines. Conditions have changed in that maximum speeds, in general, are now more than 8 per cent in excess of those common before the war. Moreover, cruising speeds are something like 10 per cent higher than they were previously.

These advances have been made possible by improved suspension, giving better road holding and comfort at high speeds; by superior steering and stability characteristics; and by better acceleration. As a result of these developments, the present day damper operates at a higher frequency and at larger amplitudes than hitherto. This gives rise to, among other things, heat dissipation problems, which are aggravated by modern trends of styling. The amount of energy dissipated by dampers currently used is estimated to be about twice as great as that dealt with by their pre-war counterparts. Indeed, it has even been claimed that on a modern fast car travelling over rough terrain, as much as 8 h.p. is dissipated by the four dampers acting together.

This may or may not be true. However, the fact cannot be denied that both the hydraulic and the friction damper methods of controlling suspension oscillations are in some respects wasteful. The power dissipated can only come from the engine, and it would be better put to a more useful purpose. Moreover, any force applied to these types of damper is reacted directly on the sprung mass, and therefore to a small extent defeats the object of employing a suspension spring.

The question that then arises is whether there is any more efficient way of damping suspension systems. There is one very promising line of approach to the problem, namely, the application of the dynamic absorber. However, before this is examined in detail, the subject of what is the exact function of the suspension system and damper must be discussed.

Function of the suspension system

The suspension system is a mechanism for isolating the portion of the chassis and body that comprises the sprung mass from vibrations caused by road surface irregularities. These vibrations may be of three types: oscillations of the sprung mass at its natural frequencies on the suspension springs, that is, bouncing, pitching and rolling; vibration induced by wheel hop, or the oscillation of the unsprung mass on the appropriate suspension spring; and irregular motions induced by the road surface contour. The tyres are in effect springs, and therefore have an influence on the natural frequencies of the different elements of the system. This influence is small so far as the sprung mass is concerned, since the tyre rate is relatively high by comparison with that of the suspension spring rate, and the system is equivalent to a mass on two springs in series. Frequencies of between about 60 and 110 cycles/min are generally adopted in practice, and the front end frequency is usually appreciably lower than that at the rear.

Wheel hop is a rather more complex phenomenon: the frequency remains constant so long as the wheel does not leave the ground, but when the wheel does hop clear of the ground the frequency becomes slower and varies with the amplitude. According to Bastow[1], the constant frequency of the first stage is given by:

$$f = \frac{1}{2\pi}\sqrt{\frac{g(t+s)}{w}} \quad \ldots\ldots (1)$$

where $g = 32{\cdot}2$, and w = the weight of the unsprung mass, and t and s are respectively the rates of the tyre and the spring.

Once the wheel has left the ground, it is subject only to the restoring force of the suspension spring. The method of calculating the frequency of this type of motion is given by Bastow as follows. Simple harmonic motion is assumed to take place, but in two separate parts. The first part is of such a form that the velocity is equal to $r\omega \sin\phi$, where ϕ is the angle measured between the point of maximum downward displacement of the wheel and the point at which the tyre is on the point of leaving the ground, and r = the maximum tyre deflection. Thus $(2\pi - 2\phi)/2\pi$ is the proportion of the cycle during which the tyre is touching the ground. The time taken by this part of the motion is

$$\frac{2\pi - 2\phi}{2\pi} \times 2\pi\sqrt{\frac{w}{(t+s)g}} = (2\pi - 2\phi)\sqrt{\frac{w}{(t+s)g}}$$

When the wheel is on the point of leaving the ground, the vertical velocity will be $r\omega \sin\phi$, and this will be true also of the beginning of that part of the cycle during which the wheel is off the ground. For this latter portion of the cycle, θ deg on each side of the point of maximum displacement, the movement corresponds to the projection of a point moving round a circle of radius R with uniform angular velocity Ω.

At the point of maximum displacement, equating accelerations gives:

$$R\Omega^2 = \frac{Rsg}{w}$$

$$\Omega = \sqrt{\frac{sg}{w}}$$

Then at the point where the wheel leaves the ground, the restoring force is $(t+s)r\cos\phi$, and is equal to $sR\cos\theta$.

Therefore, $$sR\cos\theta = (t+s)r\cos\phi$$

$$R = \frac{(t+s)r\cos\phi}{s \times \cos\theta}$$

At the same time, the velocity is $r\omega \sin\phi$ and is also equal to $R\Omega \sin\theta$.

Therefore, $$R\Omega\sin\theta = r\omega\sin\phi$$

or $$R\Omega = \frac{r\omega\sin\phi}{\sin\theta}$$

Substituting the values of R, Ω and ω already found, we have:

$$\frac{(t+s)r\cos\phi}{s\times\cos\theta}\sqrt{\frac{sg}{w}} = \frac{r\sin\phi}{\sin\theta}\sqrt{\frac{(t+s)g}{w}}$$

from which $$\frac{\tan\theta}{\tan\phi} = \sqrt{\frac{s}{t+s}}$$

The time during which the wheel is off the ground is :

$$\frac{2\theta}{\Omega} = \frac{2\theta}{\sqrt{\frac{sg}{w}}} = 2\theta\sqrt{\frac{w}{sg}}$$

and therefore the total time of the cycle is :

$$(2\pi - 2\phi)\sqrt{\frac{w}{(t+s)g}} + 2\theta\sqrt{\frac{w}{sg}}$$

or

$$t = 2\sqrt{\frac{w}{g}}\left[\frac{\pi - \phi}{\sqrt{t+s}} + \frac{\theta}{\sqrt{s}}\right] \quad \ldots\ldots\ldots (2)$$

From this the frequency can be found. In practice, the method used to calculate the frequency for different amplitudes of motion is first to choose successively increasing values of r between the point where the wheel just leaves the ground and that at which it reaches the full bump position. Then for each value of r the angles ϕ and θ are found, and they are substituted in the formula to find the time for the completion of one cycle. The frequency is, of course, equal to 1/t, and the lowest frequency is obtained with the value of r which gives an amplitude of wheel motion equal to that of full bump.

In order to isolate a mass from vibratory influences it must be mounted on springs in such a manner that its natural frequency is not more than half that of the forcing frequency, and preferably lower. This is illustrated in Fig. 1, from which it can be seen that when the ratio of the forced frequency to the natural frequency is much below 1 : 1, that is with very stiff springing, the amplitude of motion of the sprung mass is almost the same as that of the exciting force. Conversely, when the ratio is about 2 : 1, the amplitude is much reduced, and at higher ratios may even approach zero. The condition in which the ratio is 1 : 1 is one of resonance, and the amplitude would be infinite but for the fact that in practice it is always limited by damping in one form or another.

From the foregoing it follows that so far as isolation of the sprung mass is concerned it is possible to design a suspension system that will be perfectly satisfactory without a damper. However, resonant conditions must also be catered for, and it is principally for this reason that dampers are fitted. Nevertheless, because of the need to allow the spring to deflect freely under the influence of road bumps, dampers are usually designed to have a stronger action on the rebound than on the bump stroke. This means that they cannot work as efficiently as would be possible if both the bump and rebound strokes could be fully damped.

The dynamic absorber

The principle of the dynamic absorber was first put to practical use by Frahm in 1909. As now conceived for application to suspension systems, it is simply a small spring-mass system tuned in such a manner that it vibrates in antiphase to, and therefore opposing an exciting force operating at a specified frequency. Experiments have also been carried out with a flywheel type of absorber, and for other applications, different arrangements such as the pendulum type of unit may be employed.

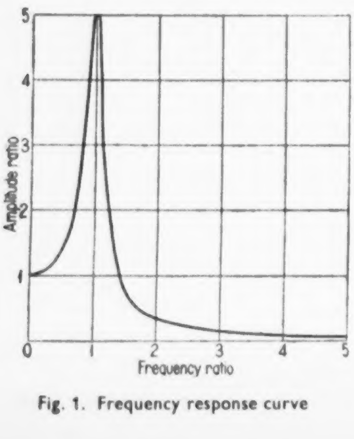


Fig. 1. Frequency response curve

There are several advantages inherent in the spring-mass system. One is that it is relatively simple, and can therefore be made inexpensively. Another is that it is mounted on the unsprung mass in such a way that there is no direct mechanical contact between the absorber and the sprung mass, as there is with conventional forms of damper. This arrangement, however, has a drawback in that the weight of the unsprung mass is increased. Such an increase will reduce the natural frequency of the unsprung mass on the suspension spring, relative to that of the sprung mass on the same spring. As a result, the transmissibility of the suspension system is increased, so that there will be a tendency towards a greater proportion of the motion of the unsprung mass being felt in the car, and the amplitude of the wheel hop motion will tend to be greater. However, both these tendencies should be more than offset by the better controlling characteristics of this type of unit.

A further good feature of the dynamic absorber is that the amount of energy that has to be dissipated as heat in the damping process is much less than with conventional units. The dynamic absorber only responds to those frequencies to which it has been tuned. In consequence, it in no way impedes the normal motion of the suspension system under the influence of random road irregularities. However, it does prevent the development of undue amplitudes of wheel hop or any other characteristic vibrations to which it may have been tuned.

There are two alternative dynamic absorber arrangements. One is the undamped and the other is the damped version. It is therefore wrong to refer to the absorber as a damper. Moreover, it does not damp the motion, but sets up a periodic force that opposes the exciting force, and under certain circumstances must itself be damped. The undamped version is not suitable for automobile suspensions, but an understanding of its fundamental principles is a useful step towards comprehending the more complex phenomena associated with the damped type.

The undamped absorber

The principles involved have been dealt with lucidly and comprehensively by Den Hartog[2]. Briefly they are as follows. The suspension system may be represented, as in Fig. 2, by a mass M and a spring of stiffness K, and the absorber as a mass m with a spring of stiffness k mounted on mass M. A force of magnitude $P_0 \sin \omega t$ acts on M, and the deflections of M and m are respectively x_1 and x_2. The natural frequency of the absorber system, which is given by $f = \sqrt{k/m}$, is arranged so that it equals that of the suspension system.

It is a relatively simple matter to write the equations of motion as follows :

$$M\ddot{x}_1 + (K+k)x_1 - kx_2 = P_0 \sin \omega t$$
$$m\ddot{x}_2 + k(x_2 - x_1) = 0 \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

Since these two equations contain only $\ddot{x}_1$, $\ddot{x}_1$, x_1 and x_2, but not the first derivatives $\dot{x}_1$, $\dot{x}_2$ and $\cos \omega t$, the forced vibration of the system is of the form

$$x_1 = a_1 \sin \omega t$$
$$x_2 = a_2 \sin \omega t$$

On this assumption, all the terms in (3) are proportional to $\sin \omega t$. By dividing throughout by $\sin \omega t$, the differential equations become algebraic equations :

$$a_1(-M\omega^2 + K + k) - ka_2 = P_0$$
$$-ka_1 + a_2(-m\omega^2 + k) = 0 \quad \ldots\ldots (4)$$

These last two formulae may be brought into a dimensionless form in the following manner :

Let $x_{st} = P_0/K$ = the static deflection of the suspension system

$\omega_a^2 = k/m$ = the natural frequency of the absorber

$\Omega^2_n = K/M$ = the natural frequency of the suspension system

$\mu = m/M$ = the mass ratio = absorber mass/suspension mass

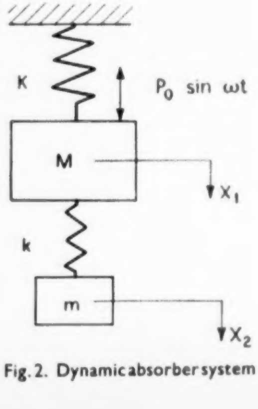


Fig. 2. Dynamic absorber system

Then by substituting these ratios in (4) and transposing we get:

$$\frac{a_1}{x_{st}}=\frac{1-\dfrac{\omega^2}{\omega^2_a}}{\left(1-\dfrac{\omega^2}{\omega^2_a}\right)\left(1+\dfrac{k}{K}-\dfrac{\omega^2}{\Omega^2_n}\right)-\dfrac{k}{K}}\quad\ldots\ldots\ldots(5)$$

and

$$\frac{a_2}{x_{st}}=\frac{1}{\left(1-\dfrac{\omega^2}{\omega^2_a}\right)\left(1+\dfrac{k}{K}-\dfrac{\omega^2}{\Omega^2_n}\right)-\dfrac{k}{K}}\quad\ldots\ldots\ldots(6)$$

From (5) it can be seen that when the numerator $1-\omega^2/\omega^2_a$ is zero, the amplitude a_1 of the suspension system mass is zero. This occurs when the frequency of the force acting on the suspension system is the same as the natural frequency of the absorber.

The next step is to determine the amplitude of motion a_2 of the absorber mass. This may be obtained from equation (6) when $\omega=\omega_a$. The first factor of the denominator then becomes zero, so that the equation reduces to:

$$a_2=\frac{-Kx_{st}}{k}=\frac{-P_o}{k}\quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots(7)$$

Thus, it can be seen from equations (5) and (6) that the main mass stands still and the damper mass vibrates with a motion $-P_o/k \sin \omega t$. Under these conditions the force in the damper spring varies as $-P_o \sin \omega t$, which is equal and opposite to the exciting force.

If $\omega_a=\Omega_n$, then $k/m=K/M$ or $k/K=m/M$. ......(8)

From these relationships it can be seen that the smaller is the absorber mass, the smaller must be its spring rate, and the larger its amplitude of motion for the spring force to counterbalance the exciting force. In most suspension systems, space considerations will limit the amplitude of motion of the absorber, and this will determine the mass ratio that must be used.

The damped absorber

It has already been stated that for application to motor vehicle suspensions the absorber must be damped. This is because the undamped type, although it suppresses vibrations of the system at the one frequency to which it is tuned, introduces two more resonant frequencies. One of these is slightly higher than the tuned frequency, while the other is slightly lower. The characteristics of this type of system are shown in Fig. 3.

If a hydraulic damper, the constant for which is denoted by the symbol c, is incorporated in the absorber mass-spring system and the main system is undamped, the equations are, for the large mass:

$$M\ddot{x}_1+Kx_1+k(x_1-x_2)+c(\dot{x}_1-\dot{x}_2)=P_o \sin \omega t\quad\ldots\ldots\ldots(9)$$

and for the small mass:

$$M\ddot{x}_2+k(x_2-x_1)+c(\dot{x}_2-\dot{x}_1)=0\quad\ldots\ldots\ldots\ldots\ldots\ldots(10)$$

By solving these two equations and substituting in the result the following symbols, a dimensionless expression for x_1/x_{st} may be obtained.

$\mu = m/M$ = mass ratio = absorber mass/main mass
$\omega^2_a = k/m$ = natural frequency of the absorber
$\Omega^2_n = K/M$ = natural frequency of the main system
$f = \omega_a/\Omega_n$ = frequency ratio (natural frequencies)
$g = \omega/\Omega_n$ = forced frequency ratio
$x_{st} = P_0/K$ = static deflection of the system
$c_c = 2m\Omega_n$ = critical damping, which is defined as the amount of damping necessary to make the system dead beat.

$$\frac{x_1}{x_{st}}=\sqrt{\frac{\left(\dfrac{2c\,g}{c_c}\right)^2+(g^2-f^2)^2}{\left(\dfrac{2c\,g}{c_c}\right)^2(g^2-1+\mu g^2)^2+\left[\mu f^2g^2-(g^2-1)(g^2-f^2)\right]^2}}\quad\ldots\ldots\ldots(11)$$

Den Hartog shows how the amplitude of motion of the main mass varies with frequency when the absorber is tuned to the forced frequency, the mass ratio is 1/20, and for different values of damping c/c_c, Fig. 4. If $c=0$, the curve is the same as Fig. 3, and when the damping is infinite, that is the absorber and main masses are locked together, the result is similar to Fig. 1. Somewhere between these two extremes of damping there is a value of c for which the lowest possible peaks of the curve are obtained.

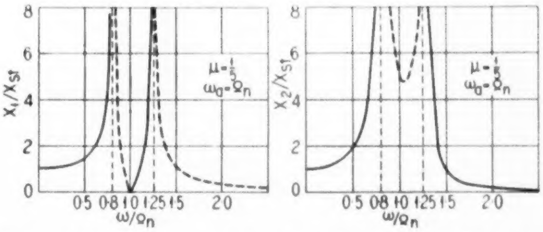


Fig. 3. Left: amplitudes x_1 of the main mass. Right: amplitudes x_2 of the absorber mass

It can be seen that all the curves in Fig. 4 pass through the two points P and Q. This is, in fact, true for all values of c. From this it follows that the least obtainable resonant amplitude occurs when the damping is such that the curve is horizontal through the higher of these two points. On this basis, calculations can be made to determine the most satisfactory degree of damping. Further improvement in the performance of the system may be obtained by altering the tuning $f=\omega_a/\Omega_n$, of the damper with respect to the main system. By this means one of the two points P and Q may be shifted up, and the other down.

The design procedure

After determining the mass ratio necessary in order that the unit will fit in the space available, the first stage in the design of a damped absorber for a suspension system is to calculate the frequency to which it must be tuned. This may be done very simply from the expression:

$$f=\frac{1}{1+\mu}\quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots(12)$$

Then the amplitude of motion of the main mass may be found from

$$\frac{x_1}{x_{st}}=\sqrt{1+\frac{2}{\mu}}\quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots(13)$$

The next step is to find the optimum damping. This could be done by differentiating (11) with respect to g, and equating to zero at the point P, but a simpler way of determining c/c_c is as follows:

Equation (11) may be simplified when g, f, and μ are fixed values and the expression becomes:

$$\frac{x_1}{x_{st}}=\sqrt{\frac{A\left(\dfrac{c}{c_c}\right)^2+B}{C\left(\dfrac{c}{c_c}\right)^2+D}}\quad\ldots\ldots\ldots\ldots\ldots\ldots(14)$$

This is independent of damping if $A/C=B/D$, and may be re-written fully as:

$$\left[\frac{1}{g^2-1+\mu g^2}\right]^2=\left[\frac{g^2-f^2}{\mu f^2g^2-(g^2-1)(g^2-f^2)}\right]^2\quad\ldots\ldots(15)$$

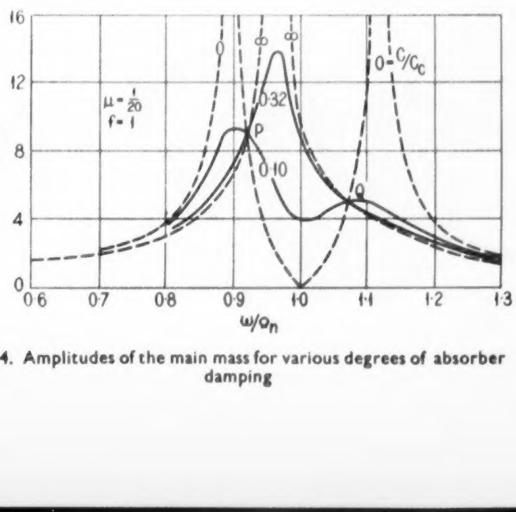


Fig. 4. Amplitudes of the main mass for various degrees of absorber damping

From this a quadratic in g^2 may be derived,

$$g^4 - 2g^2\left(\frac{1+f^2+\mu f^2}{2+\mu}\right) + \frac{2f^2}{2+\mu} = 0 \quad \ldots\ldots\ldots\ldots\ldots (16)$$

The numerical values of f and μ are known for the absorber under consideration and they may be substituted in (16). Solution of this equation gives two values for g^2. The two values, g_1 and g_2, are the horizontal co-ordinates ω/Ω_n of P and Q in Fig. 4.

Next (14) may be transposed as follows :

$$\left(\frac{c}{c_c}\right)^2 = \frac{B - D\left(\frac{x_1}{x_{st}}\right)^2}{C\left(\frac{x_1}{x_{st}}\right)^2 - A} \quad \ldots\ldots\ldots\ldots\ldots (17)$$

Now, to show how c/c_c can be evaluated it will be assumed that $g^2{}_P$ has been found to be equal to 0·71085. Then by substituting this value, together with that of x/x_{st} obtained from (13), in equation (11), the values of A, B, C and D may be calculated. If these figures were substituted in (17), the right-hand side of the expression would be zero/zero, because at the point P the damping is not determined by the amplitude. However, by substituting $g^2{}_P = 0{\cdot}7100$, which is still very close to P, and using the same value of x/x_{st}, the right-hand side of the equation (17) becomes the quotient of two very small numbers, and a definite figure may be obtained for $(c/c_c)^2$.

All that now remains to be done is to determine the relative amplitude of motion between M and m so that the stress in the absorber spring may be calculated. This may be obtained from the expression

$$\left(\frac{x_{rel}}{x_{st}}\right)^2 = \frac{x_1}{x_{st}} \times \frac{1}{2\mu g\,(c/c_c)} \quad \ldots\ldots\ldots\ldots\ldots (18)$$

In practice it is often found that the relative amplitude is too large, and the absorber spring cannot be made strong enough. However, this and other difficulties evidently can be solved, for in at least one mass produced car currently in production, the Citroen 2CV, a dynamic absorber is successfully employed on the suspension system.

The Citroen 2CV

Apart from the dynamic absorber, the Citroen suspension system, Fig. 5, is also somewhat unconventional, and it does not necessarily follow that a similar absorber could be used with equal success to replace the conventional damper in a more orthodox suspension system. It is always true to say that, in the design process, a suspension system must be considered as an entity required to match a particular vehicle layout, and not as a collection of individual components having little or no relation to one another.

Fig. 6. A view of a sectioned pivot bearing and damper assembly

Not much authoritative technical information is available concerning this car. Nevertheless, it is interesting to examine the installation, in the light of the fundamentals already explained, to see how the various problems associated with this type of unit have been overcome. Experience of the riding characteristics of the vehicle shows a general agreement between the conclusions drawn from theoretical considerations and what actually happens in practice.

The suspension system is based on four bell crank levers mounted one on each end of two tubular cross-members bolted down on top of the frame. Each lever has one long arm which, on the front pair extends forwards and on the rear pair backwards, to carry the wheels; and a second arm which is much shorter and extends downwards making an angle with the long one of appreciably more than 90 deg. The two short members on each side are interconnected, and the main suspension coil springs are incorporated, two on each side, in the interconnecting link. Thus, when the front wheels ride over a bump, the motion is transmitted through the bell cranks and links to raise the rear end and so reduce pitching. One more feature has been incorporated to complete the system ; that is a two spring arrangement to counteract any tendency for the sprung mass to settle down on one end or the other because of unequal weight distribution between the front and rear.

All the springs are assembled together in one unit on each side, Fig. 7. This unit consists of a large diameter, tubular housing in which are the two coil type main suspension

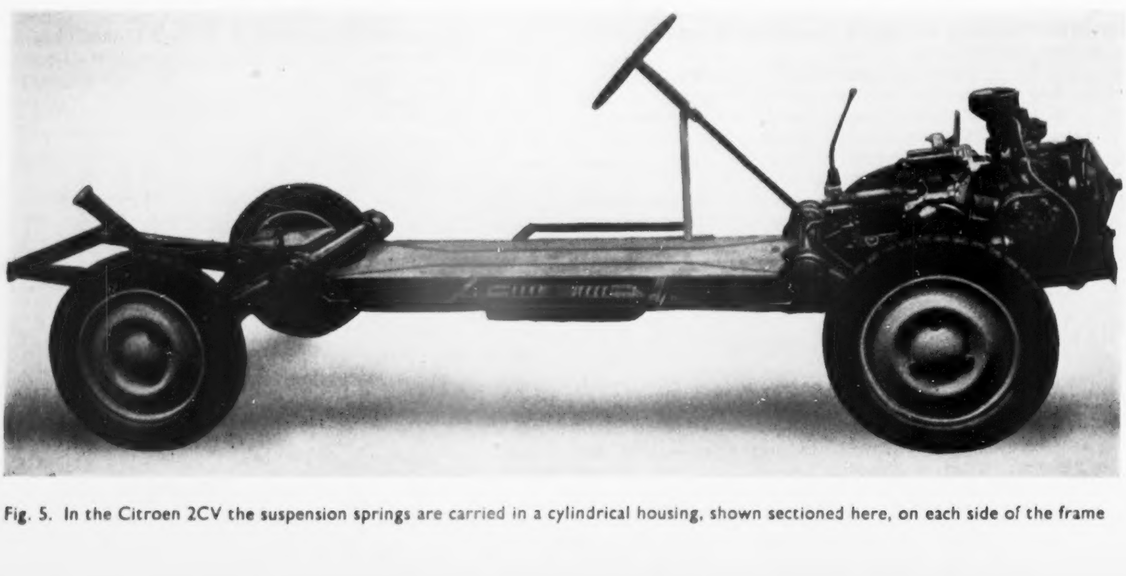


Fig. 5. In the Citroen 2CV the suspension springs are carried in a cylindrical housing, shown sectioned here, on each side of the frame

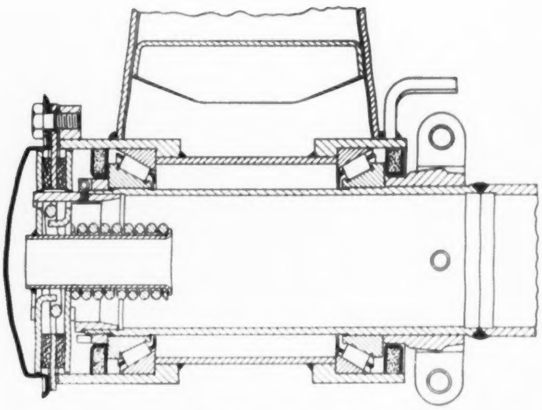


Fig. 8. Arrangement of a pivot bearing and friction damper

springs, and in the outwardly cupped end plates of which are the two conical, volute type centralizing springs. All springs are in compression. The tie rods interconnecting the front and rear bell cranks are passed through holes in the centre of the end plates of the cylinder, and their inner ends are each secured to a piston type guide which seats on the inner end of one of the main suspension coil springs. The outer ends of these two coil springs bear against the end plates of the housing. The apex of each centralizing spring bears against a shoulder on a tube mounted on a bracket on the frame side member. This tube extends through the hole in the end plate to locate the whole assembly radially.

With this arrangement any tendency to pitch is probably countered at least as effectively if not more so than by the dampers employed in more conventional layouts. Friction dampers, Figs. 6 and 8, at the pivots of the bell cranks further assist in reducing pitching, but it would appear that the primary function of these dampers is to reduce the amplitudes of the low frequency vertical oscillations of the sprung mass. Resonant vibrations of the sprung mass are further discouraged by the variable rate obtained with the bell crank geometry. Wheel hop is almost entirely countered by dynamic absorbers mounted vertically one on each wheel hub.

Each of these absorbers, Fig. 9, is in a sealed tubular housing, about $11\frac{1}{4}$ in long by $3\frac{1}{8}$ in outside diameter. A false base is pressed on the lower end of the housing to protect it against damage when passing over exceptionally rough terrain. Inside the unit, a simple mass spring system is installed. The lower end of the spring is carried in a ring, the inner periphery of which is grooved to receive it. The outer periphery of the ring is flanged and clamped between the bottom of the tubular housing and its end cap.

The mass is rather more than 5 in long. Its upper end is coned to fit in the domed top cap when at the upper extreme of its travel. For about $1\frac{7}{8}$ in below the top, the mass is a running fit in the cylinder. This portion is relieved by an annular space around it about $\frac{3}{8}$ in wide. A groove for a piston ring, the principal function of which is to centralize the mass, is cut in the centre of the base of the annular space. It would appear that this space is incorporated to reduce the area of contact and therefore the drag between the mass and its housing.

The lower portion of the mass is approximately $2\frac{1}{8}$ in diameter and extends into the coils of the spring. The top $1\frac{1}{2}$ coils are carried in grooves round a shoulder just below the part of the mass that is a running fit in the housing. A hole, about $\frac{11}{32}$ in diameter, is drilled axially from top to bottom of the mass. Passing through this hole, and extending from top to bottom of the casing, is a $\frac{7}{32}$ in diameter tube. This tube is mounted at its lower end on a tapered peg spigoted and peened in the centre of the lower cap; its upper end is located by a conical end set bolt in a boss at the centre of the upper cap. Two holes are drilled diametrically through the tube, one at the lower end and one near the upper end.

It seems that damping is afforded by the passage of air, as it is displaced from one side of the mass to the other, along the clearance between the hole in the mass and the tube. A small quantity of oil is carried in the base of the unit so that air pressure during the downward stroke of the mass forces the oil in the lower hole, up the tube and out of the top hole, from which it is sprayed on to the conical end of the mass, and runs on to the walls of the housing for lubrication purposes.

Because of the lack of information on this unit, it is only possible to estimate very roughly as to how well it satisfies the theoretical requirements. Although the unsprung mass of the vehicle appears to be small, it is doubtful whether it is any less than that of a more conventional wishbone suspension system for such a light car. This means that the ratio of the absorber mass to unsprung mass is probably fairly high, and the length of travel available may not be quite enough for the balancing out of very large amplitudes of wheel hop. However, since such amplitudes are usually built up from smaller ones by a succession of appropriately timed impacts, the length of travel available may be adequate because it is great enough to allow the absorption of smaller amplitudes of wheel hop, thereby preventing build up. Wide rimmed wheels and large section tyres are used to ensure that the road surface irregularities can be traversed without inducing large vertical motions of the wheels.

An interesting feature of this absorber is that it has a variable rate spring because of the restriction of the passage of air from one side of the mass to the other. Thus, for small amplitudes of motion, when the velocity of the mass will be small, the air will only add slightly to the spring rate. On the other hand, for large amplitudes of motion, the velocity

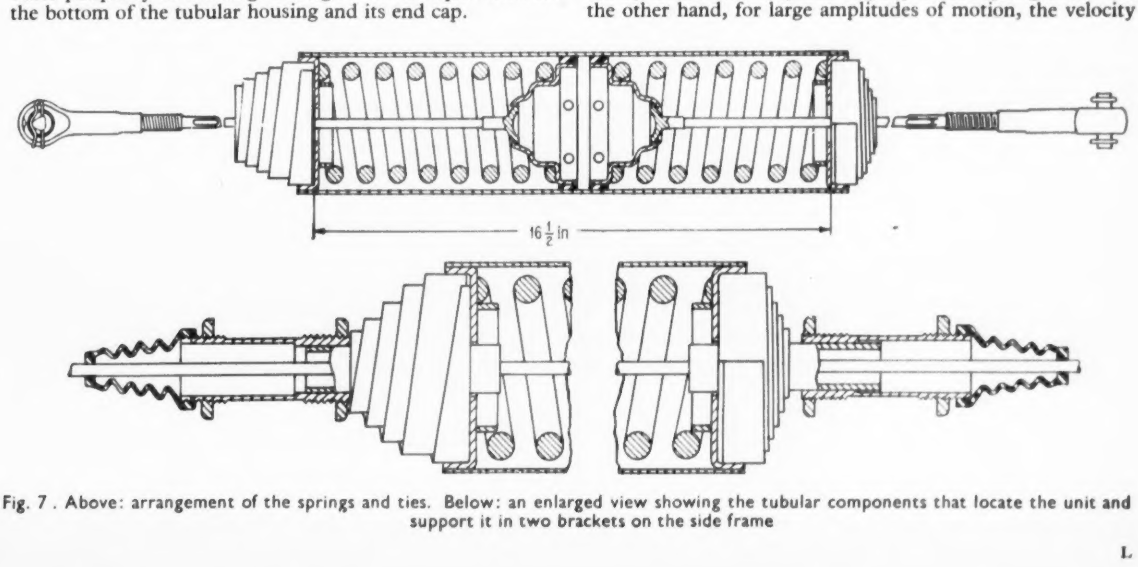


Fig. 7. Above: arrangement of the springs and ties. Below: an enlarged view showing the tubular components that locate the unit and support it in two brackets on the side frame

of the mass will be greater and the air will be compressed because there will not be time for its passage through the central hole. Therefore, the effective spring rate will be higher since it is the sum of the effects of the coil spring and of the compressed air. This will have the effect of increasing the natural frequency of the absorber mass system under conditions when, if the wheel is hopping clear of the ground, the natural frequency to be absorbed is decreasing. From theoretical considerations it would appear that in order to maintain a constant tuned relationship between the two frequencies, the absorber mass frequency should be decreased rather than increased as the amplitude of the wheel hop increases.

However, the system certainly works very well in practice. When driven over extremely rough terrain, the vehicle bounces to an extent which is somewhat frightening to anyone accustomed to more conventional systems. The bounce is, of course, at the fairly low natural frequency of the sprung mass on the suspension. The vehicle also rolls when cornering. Despite these characteristics, the Citroen 2CV is perfectly safe, and the rolling and bouncing behaviour probably tends to discourage abusive handling of such a small car. Whether the same suspension system would be suitable for a faster and larger car is questionable. The most outstanding feature of the suspension system unit is the almost complete absence of wheel hop. Because of this, the steering and road holding qualities over rough country or pavé are exceptionally good.

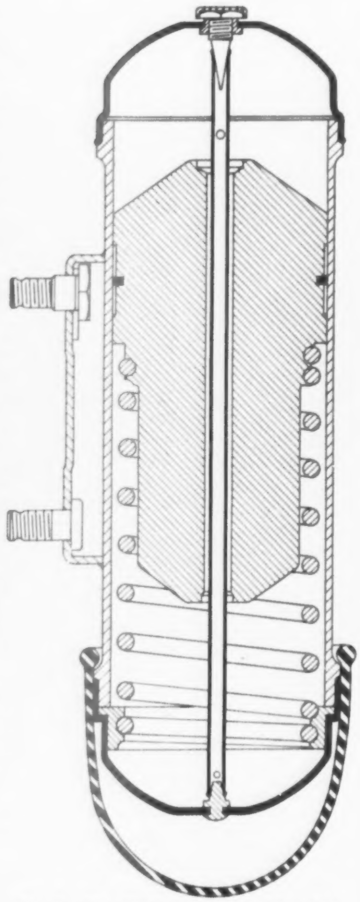


Fig. 9. The Citroen dynamic absorber

Other systems

Other forms of dynamic absorber can be employed, but none has so far been put into production for use on motor vehicle suspensions. It is interesting to note in passing that a simple spring-mass system, called l'Autostable, with its axis positioned laterally at the rear of a vehicle, has been marketed by a Swiss company. A claim is made that it prevents snaking, a defect sometimes experienced particularly in racing and sports cars. Other systems that might be used are torsional pendulum types.

In the discussion that followed a paper entitled "Shock Absorbers," by J. W. Kinchin and C. R. Stock, read recently in London at the Automobile Division of the Institution of Mechanical Engineers, Mr. H. C. Barty, of Spooner Motors Ltd., Gravesend, described a system with which he was experimenting. In this system, the inertia of an oscillating flywheel, appropriately phased, was used to oppose the forces set up in the suspension under resonant conditions.

In order that the flywheel might be as small as possible, it had to oscillate at high speeds. This was done by mounting it on the input shaft of a steering box carried on the back axle, and connecting the end of the drop arm to the vehicle structure, Fig. 10. Relative motion between the axle and the chassis caused movement of the drop arm. With this arrangement the steering box was working in the reverse direction to that for which it was designed, so the mechanical friction must have been fairly high. This friction provided the damping force, which was almost directly proportional to the applied load, so that if it was of the right magnitude, it no doubt satisfied the theoretical requirements.

The timing of the mechanism so that its motion was in antiphase to that of the suspension was effected by interposing a flexible link between the drop arm and the chassis. It can be seen from the illustration that the drop arm was more or less horizontal when in the static position, while the connection to the frame was approximately vertical and offset an inch or so from the end of the drop arm. The flexibility of the link, which was of steel, was provided by fitting rubber bushes to carry the lower end of the connection to the chassis and the outer end of the drop arm. The torsional resilience of these bushes was such that the spring rate of the link system was the equivalent of 15-20 per cent of the vertical rate of the suspension springs. In the particular installation shown in the illustration, the steering box was mounted on a rolled steel section bolted to the axle, because in the early stages of the investigation it was considered advisable to avoid applying additional loads to the centre of the axle. However, it was found later that this was an unnecessary precaution.

The system operates as follows. In the event of a rapid upwards displacement of the axle, the rubber bushes in the links connecting the drop arm to the frame deflect torsionally, and a force is applied to the end of the drop arm. This force overcomes the friction of the steering gear and the inertia of the flywheel, which then rotates at an increasing speed. In the meantime, the axle will have fallen again to a level below the normal static position. Again, the initial deflection is allowed for by the torsional resilience of the rubber bushes. The load to the drop arm is reversed, but the inertia of the flywheel, operating through a favourable gear ratio, opposes that load. This causes the flywheel to slow down and stop, and its direction of rotation is reversed, so that it then opposes upwards motion of the axle. This cycle of operations is repeated until the oscillation of the sprung mass ceases.

This system, because of the gearing and the linkwork involved, is more expensive than the simple spring-mass absorber. It would appear that as used in the experiments described, the absorber was tuned to the relatively low frequency of the sprung mass on the suspension springs, and that it can hardly have had much effect on wheel hop.

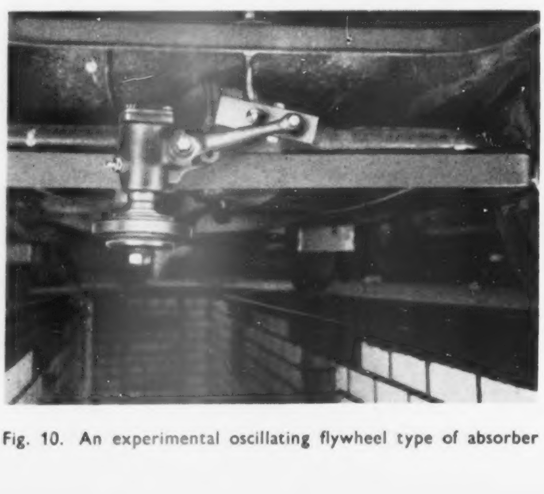


Fig. 10. An experimental oscillating flywheel type of absorber

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OIL GROOVING

The Trojan Universal Machine

SOME time ago Trojan Limited, Purley Way, Croydon, received a large contract for phosphor bronze bushes which required both right- and left-hand multiple oil grooves. There was no existing machine on which the oil grooves could be cut economically and Trojan Limited decided to explore the possibility of developing a special machine. While the machine was primarily needed for carrying out the phosphor bronze bush contract, it was considered advisable to investigate the possibility of developing a machine that would be truly universal within its capacity. That is, a machine that would cut all types of oil grooves either individually or for quantity production. The machine as finally developed is shown in Fig. 1.

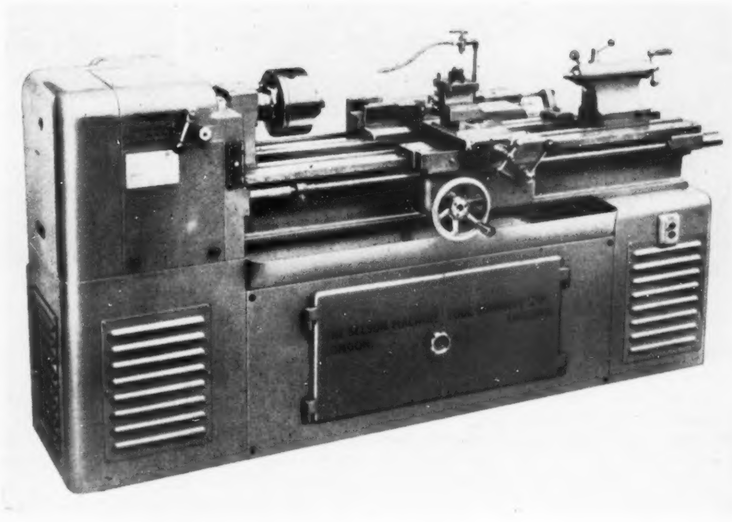


Fig. 1. The Trojan universal oil grooving machine

The headstock spindle is motor driven through a vee belt and worm gearing. By disengagement of a hand-operated dog clutch it can be indexed to any divisor of 24. To allow any number of oil grooves to be generated in a continuous cut, the spindle is coupled to a motion crank through change gears. The circular crank motion is changed to a reciprocating motion for the saddle stroke through a cross-head. Provision is made for adjustment of the saddle stroke length. The saddle is locked to the reciprocating shaft by a handle and is maintained in position by a locking stop collar.

A cam under the saddle relieves the tool on either the forward or backward stroke to allow right- or left-hand grooves to be cut. Special cams can be fitted to cut eccentric face grooves, and a tailstock can be used for long external shafts. By disengaging the clutch and allowing a plunger to move between the driving dogs, longitudinal grooves can be cut to any divisor of 24 with the spindle stationary. For taper grooving, a removable sub-slide and an angle control plate are clamped in grooving position by means of an anchor bracket. Some of the many types of grooves that can be cut on this machine are shown in Fig. 2. All standard oil grooves normally required on outside diameters, on inside diameters, on external and internal tapers and on the faces of components can be cut.

Multiple grooves of any number of equal spacings and any helix angle or lead within the capacity of the machine can be generated in a continous cut without stopping the machine for spindle indexing. This is effected by mounting the appropriate slip change gears to allow the tool to reciprocate the desired number of times to one revolution of the workpiece. For diamond grooving, there is continuous cutting on both the forward and backward strokes of the tool.

Obviously, the generating process shows a remarkable saving in time as compared with the method hitherto used for multiple oil grooves, in which each groove had to be cut singly and the spindle had to be stopped and indexed between one groove and the next. Furthermore, the quality of the work is better. The helix produced by the generating process is a true harmonic curve on each side of the centre line of the groove. This is obtained by the use of a cross head instead of the more usual connecting rod. The machine set-up for generating multiple diamond grooves is shown in Fig. 3.

Eccentric face grooves may be cut up to a limit of approximately 1 in eccentricity by mounting a suitable cam in place of the standard cut relieving cam. It is also possible to cut broken grooves. For this application where the groove is cut in only parts of the helix, the standard cut-relieving cam is replaced by a special cam that moves the tool away from the work during the appropriate parts of the stroke.

For taper grooving with an attachment, the crank stroke is set to length and the saddle to the correct position for the required groove. The cam follower bar is then adjusted to the full length of its travel away from the relieving cam, the angle control plate is adjusted to the zero position, and the taper grooving slide is positioned between its ways. The taper turning slide is clamped to the bed by means of the anchor bracket, in a position central to the mean of the saddle stroke.

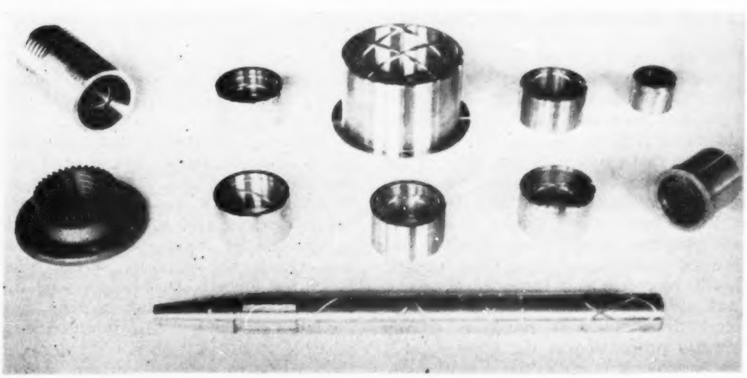


Fig. 2. Examples of grooves cut on the Trojan machine

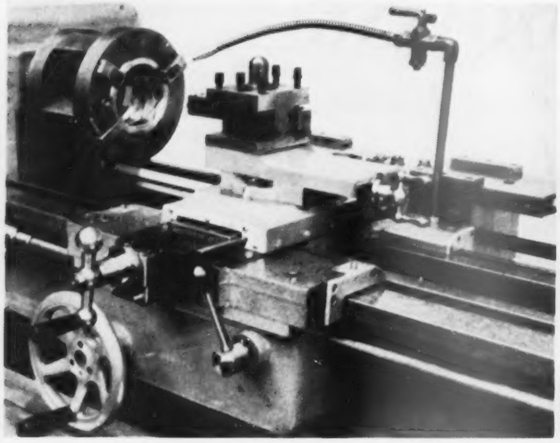


Fig. 3. Machine set up for generating diamond grooves

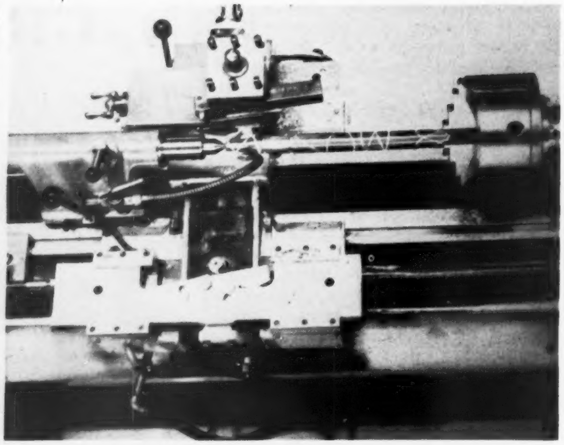


Fig. 4. Set-up for producing grooves on an external taper

After the angle control plate has been locked at half the included angle of the required taper, the roller follower is screwed into position on the cam follower bar. The angle control plate actuates the cross slide by engaging the roller follower. A set-up for external taper grooving is shown in Fig. 4.

Although this machine is essentially intended for use in cutting oil grooves, it can also be used for cutting accurate internal and external splines to any divisor of 24. The machine is driven by a $1\frac{1}{2}$ h.p. motor and there are six spindle speeds from 49 to 160 r.p.m. It will cut grooves up to a maximum length of 7 in. The maximum diameter of work that will be admitted over the saddle and bed is 14 in and over the cross slide 8 in. Work up to a maximum length of 46 in can be accommodated between centres. The Selson Machine Tool Company Ltd., Cunard Works, Chase Road, North Acton, London, N.W.10, are the sole agents for the Trojan universal oil grooving machine.

VALVE SEAT DISTORTION

A PAPER by J. A. Newton and M. J. Tauschek, entitled "Valve Seat Distortion," has been published in an S.A.E. Preprint, March 3-5, 1953. The effects of valve seat distortion, its causes and methods of minimizing it are discussed. Seat distortion, whether thermal or mechanical, is now one of the chief types of valve malfunctioning. The distortion may be detected by the presence of burnished areas on some portions of the seating surface while other areas are dulled by deposits. Localized areas of leakage result in valve temperature rise and valve burning. In faced valves, the face cracks and in addition severe guttering occurs behind the surface.

Distortion of the valve seat may be caused by excessive temperature of the cooling water in the vicinity of the valve. This was revealed upon examination of a unit in which there had been a particularly high incidence of valve failures. Thermocouple measurements made in 60 different positions in the cylinder head proved that, although the average coolant temperature was low enough for adequate cooling, the temperature in passages between exhaust valve seats was 270 deg F, that is about 10 deg F above the boiling point for this system operating under pressure.

If distortion has been experienced, the surest method of minimizing it is by redesigning the cylinder head to open up the critical passages. Alternatively, excessive local temperatures can sometimes be lowered by means of properly situated coolant flow control tubes. Other measures that may be adopted include the use of faced valves, faced and coated valves, sodium-cooled valves, soft valve seat insert materials, a flexible insert which allows for some distortion of the cylinder head and, above all, the use of a positive valve rotator. *M.I.R.A. Abstract No.* 6274.

THE ENGINE BEARING

IN an article entitled "The Engine Bearing", in the Journal of the Institute of Automotive and Aeronautical Engineers, November, 1952, the author, D. G. Soutar, states that the basic requirements of an engine bearing are to absorb combustion shocks, and to control the conversion of reciprocating movement into rotary movement. Moreover, it must resist wear and must not cause wear of mating parts. It must also perform satisfactorily when lubricated with contaminated oil, and at fairly high temperatures.

Most of these requirements are satisfied by suitable choice of material. Seven main qualities are important: they are, load-carrying capacity, fatigue resistance, conformability, embeddability, corrosion resistance, low wear rate, and economy. A series of polar diagrams is given for the better known bearing materials, in which each of the above seven qualities is plotted along a separate radius. The lead base and tin base babbits are shown to excel in all qualities except load capacity and fatigue resistance, while the copper leads are excellent in these two respects, but not so good in the others. Copper lead silver alloy is rated as having similar properties to the copper leads except that it is much more expensive. It is pointed out that the wear rate, corrosion resistance, embeddability and conformability of the copper lead alloys can be greatly increased by plating with a lead tin alloy of 0·001 to 0·005 in thickness.

The effect of extreme load on bearing life is also discussed, and it is shown how the bearing load due to inertia and centrifugal force at the end of the exhaust stroke can greatly exceed the firing load at high speeds, while at low speeds the opposite is the case. Since the former loading comes on the cap-half of the big end bearing, and the latter on the rod half, it is generally possible to distinguish readily between failure due to excessive engine speeds and failure due to labouring at low speeds. *M.I.R.A. Abstract No.* 6242.

DIESEL ENGINE EXHAUST

Developments for Accurate Evaluation of Appearance

A. L. Wachal, Dipl.Ing., A.M.I.Mech.E., M.Inst.Pet.

THE phenomenon of exhaust smoke occurring in diesel engine operation is a problem which is still unsolved and is the subject of research work both in the automotive and petroleum Industries. This unfortunately familiar characteristic of the diesel engine unfavourably influences the general public towards usage of diesel engine in city and general road transport; even the law (Road Traffic Act of 1930) is not indifferent towards "emitted smoke, visible vapour etc." from road vehicles.

In enclosed spaces such as mines, factories, etc., the diesel smoke is even less welcomed. The petrol engine, though generally working with much lower fuel economy than the diesel engine and often producing highly toxic exhaust gas, is readily accepted on the road only because it cannot easily be realized how much the air may be polluted by incompletely burnt petrol. In this respect the diesel engine has all grounds for claiming superiority over petrol engines.

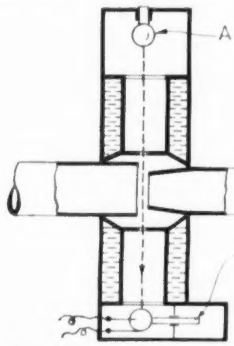


Fig. 1. Diagrammatic arrangement of a full-flow smokemeter

A great deal of research effort has already been devoted to the worthwhile task of reducing exhaust smoke. It seems, however, that the problems involved are by no means simple and that further extensive and very systematic study is needed before it is learnt how diesel smoke originates and whether there is still any available means to offer for its more effective control.

To enable such an investigation to be conducted methodically it is of primary importance to adopt a convenient and accurate method of smoke rating by which small changes in the smoke density can be observed.

In this paper some typical designs of smokemeter are surveyed and a new design is described in which an attempt is made to alleviate disadvantages of existing smokemeters. Apart from the most popular but the least accurate visual valuation, two methods have been exploited in rating the engine exhaust smoke.

The first method to be considered adopts various means of filtration for the removal of all particles inherent to the dark appearance of the engine exhaust. The quantity of the removed deposit is usually determined gravimetrically or indirectly by quantitative measurement of carbon dioxide produced from the complete combustion of the collected deposit. The smoke density is then expressed in terms of free carbon content in a unit weight of the engine exhaust. This method, although accurate and widely applicable, has a very limited value in the field, mainly because of the tedious work involved in the filtration and the measuring of the volume of hot engine exhaust; it is also prone to experimental errors in the determination of the relatively small quantities of carbonaceous deposit.

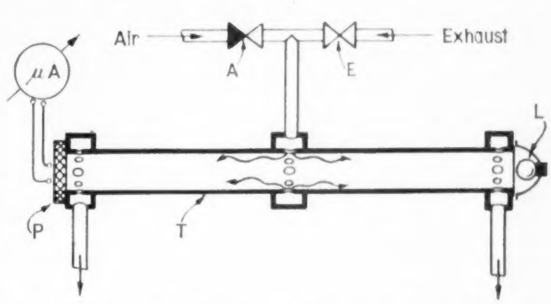


Fig. 2. Diagrammatic arrangement of a simple sampling smokemeter

The second method widely used for the assessment of engine exhaust capacity comprises a photo-electric cell and a constant source of light separated by a column of exhaust gas. This principle is used in two distinct types of photo-electric smokemeters; one which could be described as a full-flow and another as a sampling-type apparatus.

The "full-flow" smokemeter probably evolved from its simple prototype in which the light source and the photo-cell were placed opposite in the chimney stack. In this arrangement by measurement of the photo-electric current, it is possible to detect any undesirable increase in contamination of industrial waste gases. Smokemeters designed on the same principles for engine exhaust systems usually suffer from overheating and fouling by soot of the cell and the light source; and from difficulties in checking the clear exhaust datum point while the engine is working. There have been a great number of attempts to overcome these difficulties and some success has been met.

An ingenious design of the full-flow smokemeter which, it is alleged, deals successfully with the above difficulties is described in detail by A. Bokemueller (1) and is shown diagrammatically in Fig. 1. The ejector-like arrangement of the exhaust pipe inside the body of the apparatus leaves a gap through which the light passes before reaching the photo-cell, and the suction effect prevents the sooty exhaust from impinging on the cell and electric lamp. The water jacket around the lateral extensions of the meter keeps its essential optical parts at a controlled temperature. To account for changes

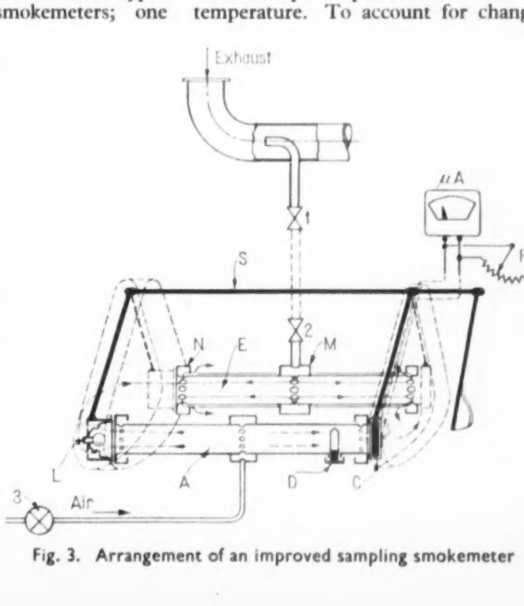


Fig. 3. Arrangement of an improved sampling smokemeter

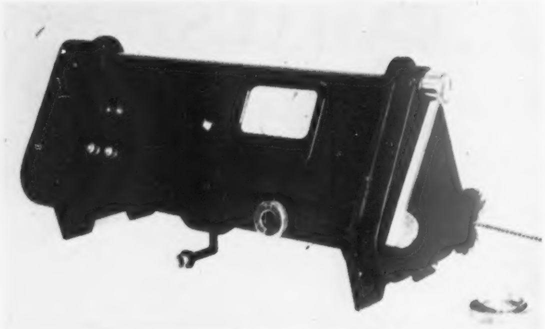


Fig. 4. The instrument shown schematically in Fig. 3

in the light intensity of the electric bulb a special compensating system with an additional photo-cell has been incorporated.

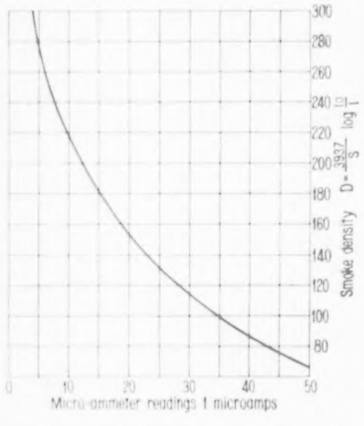
Other designs use compressed air for screening the photo-cell and light source (2); and a retractable air-filled tube inside the exhaust gas chamber has also been used for checking the no-smoke datum. The most important disadvantage, however, of this type of smokemeter is that it is not easily adaptable to existing engine installations, and the readings cannot simply be compared when the same apparatus is used with engines of various sizes. The sampling type smokemeter is more convenient in this respect. It is usually built as a portable instrument so that is can be used wherever desired. Its numerous designs vary only a little from one another and have been described in several earlier publications (3, 4, 5, 6, 7 & 8); the general principle is shown diagrammatically in Fig. 2. The electric bulb of suitable power (L) and a barrier layer type photo-cell (P) are held at opposite ends of a metal tube (T) into which a sample of exhaust gas and clean air are introduced in turn through the cocks (E) and (A) respectively. By scavenging the smokemeter with the air a no-smoke condition can be obtained for which the light intensity may be adjusted to give a convenient instrument reading. When smoky exhaust is introduced in place of the air, the output of the photo-cell will be reduced according to the amount of light absorbed by the smoke.

However, owing to intermittent scavenging by air the smoke tube seldom gets a chance to warm up enough to prevent certain amounts of condensation of exhaust water occurring when the exhaust gas replaces the air. This, in turn, means fouling of the tube and the photo-cell by sooty water; probably smoke readings are also directly affected by the formation of mist and an extraction of soot when droplets of condensate settle on the walls of the apparatus.

The deposition of soot on the light source and the photo-cell is progressive and the light needs to be intensified to counteract this effect, thus causing changes in the light spectrum characteristic which can affect the smoke rating. For these reasons readings of smoke density obtained from the type of smokemeter described are inevitably of a low order of reliability. Various means of protection of the photo-cell may suggest themselves; hot compressed air, and additional heating of the exhaust sampling pipe (8), have been used to overcome these troubles; but such methods make smoke rating rather a complicated and skilful procedure.

The aim in designing the new smokemeter was to combine the advantages of both of the above types of meters while alleviating as much as possible their disadvantages. The new smokemeter is diagrammatically illustrated in Fig. 3 and in its finished form is shown in Fig. 4.

The basic differences between this and the conventional designs of sampling type smokemeter is that in the former, separate tubes are provided for the exhaust gas and the air. The photo-cell and the light source are movably mounted so as to permit their simultaneous movement from the position of facing ends of the air tube to the identical position in relation to the sample tube. Means are also provided whereby the cell and the light source, while facing the exhaust tube, are isolated from smoke by a stream of air introduced from the reference tube. In operation an appreciable portion of the exhaust gas, branched from the engine exhaust system, enters the asbestos lagged sample tube (E) via a circular muff (M) and radially drilled holes in the middle of the tube, thus ensuring an even distribution of the exhaust gas over the cross-section of the tube. Exhaust outlet ports, similar, but somewhat larger in the total area, are provided at both ends of this tube so that the gas moves outwards in both directions, forming in it an 18 in long gas column. To prevent light from the outside penetrating into the tube, the exit ports communicate with the atmosphere through a series of holes inclining towards the tube in the side of the exit muffs (N).



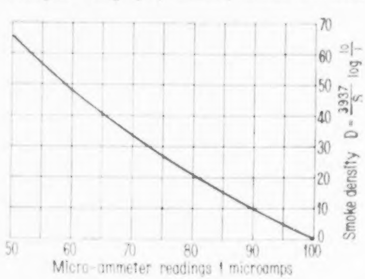
$S = 18$ in. $I_0 = 100$ microamps. $I = 0\text{-}50$ microamps.

Fig. 5. Meter readings—smoke density. Conversion graph for 18 in long smoke meter

The air is admitted to the reference tube (A) in a similar manner to the exhaust gas and flows also in both directions inside the tube. A stream reaching the light source (L) passes through holes in the reflecting mirror and is used to cool the electric bulb. This portion of air finds its way out through exit ports in the smoke tube. When the lamp is in the position facing the exhaust tube, the air passes through the same holes in the reflector but in the opposite direction, thus protecting the light source from fouling by impingement of exhaust gas. A portion of the air stream flowing in the opposite direction passes through the gap between the photo-cell and the side plate, cooling and screening the cell from direct contact with exhaust. To make this arrangement less sensitive to variation in the air inlet pressure, restricted exits for excess air are provided at both ends of the tube, similar to those described above.

The light source consists of a 6V 36W standard electric bulb in a parabolic reflector, and the position of the bulb inside the reflector is adjustable. A barrier-layer type photo-electric cell C, 45 mm. in diameter is located opposite the light source in the brass holder which embraces the ebonite frame of the cell. An adjustable screw on the one side, and the aluminium cover on the other, provide limits for the rocking angle of the cell arm, ensuring the same position of the cell in relation to the tubes when readings are taken.

Exhaust gas passing continuously through the smoke tube inevitably leaves some black deposit on its inner surface, which may slightly reduce the intensity of light reaching the cell. To compensate for this and to ensure the same photo-cell output from both tubes under no-smoke conditions, a fine adjustment for light restriction is needed in the air tube. This is provided by inserting across the light a dimmed Perspex strip (D) which, when turned,



$S = 18$ in. $I_0 = 100$ microamps. $I = 0\text{-}50$ microamps.

Fig. 6. Meter-readings—smoke density. Conversion graph for 18 in long smokemeter

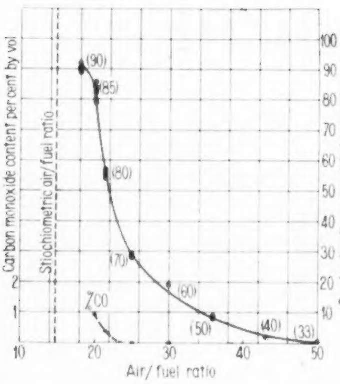


Fig. 7. Smoke density and carbon monoxide content vs. air/fuel ratio, 1,500 r.p.m. The figures in brackets indicate corresponding b.m.e.p. (p.s.i.)

reduces the light reaching the cell. To facilitate cleaning of the smokemeter tight fitting caps are provided in side-covers opposite both ends of the smoke tube.

In the layout suggested in Fig. 3, a sample of engine exhaust gas is drawn through the ½ in I.D. stainless steel tube inserted just below the elbow in the engine exhaust pipe, and pointing in the upstream direction. An effective lagging of the sampling pipe is recommended when its length exceeds about three feet. Two stop cocks allowing an unrestricted flow of the exhaust are provided, one at each end of this pipe. The stop cock (1) serves as a means of adjustment of the exhaust flow rate and the stop cock (2) placed next to the smokemeter is used for checking the no-smoke datum of the sample tube during a test. The air supply line is provided with the pressure reducing valve (3), by means of which the air inlet pressure can be suitably adjusted.

The readings obtained from the smokemeter give numerical values of the percentage reduction, "S" of light caused by the 18 in long column of engine exhaust between the light source and the photo-cell

i.e. $S \text{ per cent} = \frac{A - E}{A} \times 100$

where A = Intensity of illumination of the photo-cell when facing the air-filled tube.

E = Intensity of illumination of the photo-cell when facing the exhaust column.

If the illumination of photo-cell is kept within the range of linear response, the corresponding currents measured by the meter, I_0 and I, are proportional to the intensities of illumination, A and E, respectively.

$$\therefore\ S \text{ per cent} = \frac{I_0 - I}{I_0} \times 100$$

The length of 18 in seems to be much favoured by smokemeter designers and the percentage of light absorbed provides in many cases an adequate basis for comparing the smoke density results. In an effort, however, to make the measurement of smoke density more widely comparable many investigators prefer to follow K.M. Brown (4) in using smoke density related to the unit length of smoke column thus making it independent of the length of the smokemeter used.

The method of bringing the readings of smokemeters of different lengths to a common basis is derived from the following consideration:—

Let E = intensity of illumination of a photo-cell placed at the end of a smoke column of length s.

A = intensity of illumination of a photo-cell when the column of smoke is replaced by air.

Let $\frac{1}{a}$ represent the fraction of light absorbed per unit length of smoke column.

Now when $s = 0 \quad E = A$

$$s = 1 \quad E = A\left(1 - \frac{1}{a}\right)$$

$$s = 2 \quad E = A\left(1 - \frac{1}{a}\right)^2$$

$$\text{Thus} \quad E = A\left(1 - \frac{1}{a}\right)^s$$

$$\frac{A}{E} = \frac{1}{\left(1 - \frac{1}{a}\right)^s}$$

$$\log \frac{A}{E} = s \log \frac{1}{\left(1 - \frac{1}{a}\right)}$$

$$\frac{1}{s} \log \frac{A}{E} = \log \frac{1}{\left(1 - \frac{1}{a}\right)} = D$$

This formula for smoke density per unit length will give constant values of "D" for varying s and A. For convenient numerical values of "D", Brown (4) suggests 100 metres as the unit length of smoke column. For this length the formula for smoke density becomes:—

$$D = \frac{3937}{s} \log \frac{A}{E}$$

$$\text{or} \quad D = \frac{3937}{s} \log \frac{I_0}{I}$$

Graphs based on this formula for conversion of smokemeter readings to smoke density units are given in Figs 5. and 6.

The behaviour of the new smokemeter was examined in a series of smoke density tests. The object of these tests was to find out the extent to which the design and operation of the instrument could be blamed for experimental error in the determination of smoke density. It being difficult to produce a stable gaseous mixture of controlled transparency the tests were carried out in conjunction with two single cylinder diesel engines.

In Fig. 7 the smoke density readings, expressed as percentage of the light absorbed, are plotted against the air/fuel ratio; also the carbon monoxide content in exhaust gas is shown in this figure. For each air/fuel ratio effected by a change in the engine loading, five consecutive smokemeter readings were made at one minute intervals, and during this period samples of exhaust gas were drawn for analyses from which the air/fuel ratios were determined. It will be noted in this figure that when

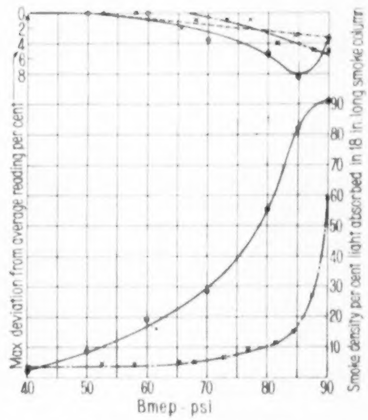


Fig. 8. Smoke density vs. b.m.e.p., 1,500 r.p.m.

the engine was run at 1,500 r.p.m. high smoke density was observed at air/fuel ratios much weaker than chemically correct and that the considerable spread in smoke readings coincided roughly with the appearance of carbon monoxide in the exhaust gas. This fluctuation in smoke density readings became less pronounced when the percentage of light absorbed in the 18 in long smoke column exceeded approximately 80. In Fig. 8 the maximum percentage deviations in smoke density readings are shown above the corresponding average smoke density values which are shown as single points on the smoke density curves.

The author wishes to express thanks to the Chairman of the Anglo-Iranian Oil Co. Ltd. for permission to publish this article.

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STROBOSCOPIC DEVELOPMENTS

Interesting New Instruments

TWO new stroboscopic devices have recently been developed by Dawe Instrument Ltd., 83, Piccadilly, London, W.1. One, the type 1206 "slow motion" stroboscope, has been specifically developed for vibration testing. Every component has a natural frequency of vibration and if excited at this frequency, resonance is set up and may, in extreme cases, lead to failure. It is therefore essential that care be taken to ensure that the natural frequency of a component is outside the range that will be encountered in service.

To ensure that this condition is met, prototypes and production models are often tested on a vibration table so that such resonances can be detected and, if necessary, remedied. If a stroboscope is flashed on the component under test at the same rate at which the component is vibrating, the motion will be frozen and the object will appear to stand still. By slowing down the flash rate slightly in relation to the frequency of vibration, the component is made to appear to vibrate in slow motion.

Dawe type 1207 mains frequency stroboscope. An inexpensive instrument for examination of phenomena in any way dependent upon the mains frequency

The Dawe type 1206 slow motion stroboscope has been developed specifically for this kind of application. It is, in fact, a special-purpose stroboscope that enables test pieces on a vibration table to be viewed in slow motion over a wide range of vibration frequencies. The advantage of slow motion over complete freezing in this type of application is that it permits the complete cycle of motion to be seen clearly irrespective of the driving frequency.

Essentially, the 1206 equipment comprises a high intensity stroboscope together with an oscillator for driving a power amplifier and vibration generator. The stroboscope can be driven at the same frequency as the oscillator or alternatively with a constant frequency of ½, 1 or 2 cycles per second irrespective of the oscillator frequency. In fact, the freezing operation is the limiting case of slowing down.

The normal procedure is to vary the frequency of the oscillator while the object on the vibration table is under observation. If the frequency difference is set to 2 cycles, the movement of the object that is being tested under the action of vibration will go through a complete cycle of movement in ½ second. For a more detailed study of the movement, the frequency difference can be switched to ½ or 1 cycle. The pattern of movement is precisely displayed and any undesirable effect will be clearly shown.

A feature of this stroboscope is the very high repetition rate of 500 c/s. which is obtained by the use of a special circuit for the flash tube. The tube is xenon-filled and gives a white light. The flash duration is approximately 40 micro-seconds. A power supply at 200/250 volts at 50/60 c/s is required.

A Dawe type 1207 mains frequency stroboscope is shown in the accompanying illustration. This is a special-purpose instrument that operates directly from the mains. It comprises a neon flash tube mounted on a compact handle which houses the necessary controls. The flash tube is automatically triggered at the mains frequency. This design eliminates much of the complication, such as the necessary triggering action, of instruments intended for more general application. As a consequence it is not only simpler but also cheaper and more robust than previous instruments.

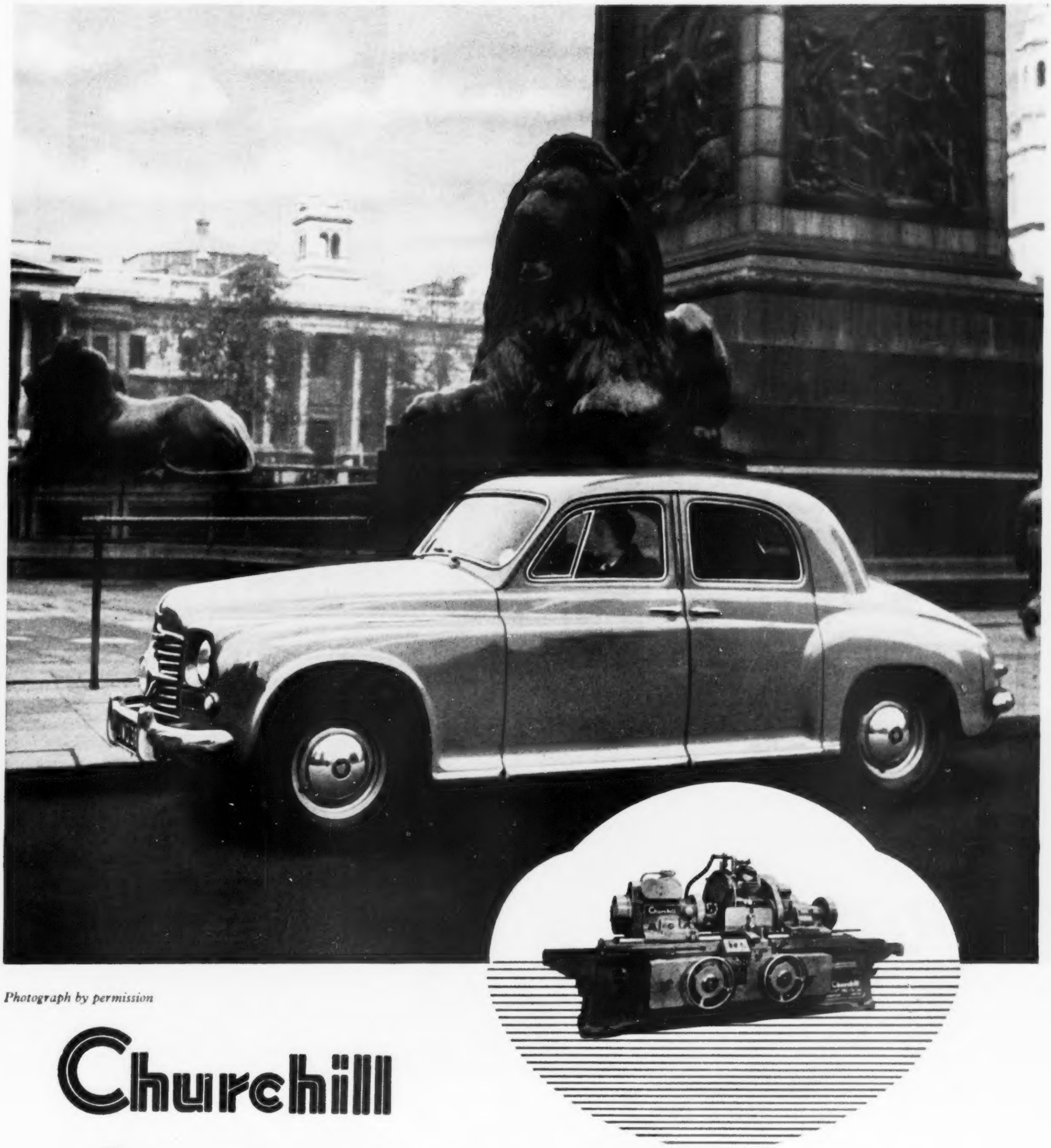
Synchronism of the flash with the mains frequency necessarily places certain limitations on the uses for which the type 1207 instrument is suitable. It is, however an ideal instrument for the examination of phenomena that are in any way dependent on the mains frequency, including multiples of the mains frequency. Typical of such applications are time switches and timing devices.

The use of this stroboscope is not, however, restricted to items having an operation exactly synchronized with the mains frequency. For example, induction motors have a small degree of slip relative to the mains synchronous speed, and when an induction motor is viewed under the light of the type 1207 stroboscope, which is plugged into the same mains, the rotor will appear to be rotating at the speed of slip. An interesting feature is that the rotor will appear to be rotating backwards. Irrespective of the direction of rotation. it is possible to count the slip directly. By loading the rotor, it is also possible to see directly the point at which the torque reaches the stalling point, On synchronous motors the point of pull-out can be tested.

The two main classes of work for which this instrument is best suited are, firstly for testing motors and certain equipment, and secondly in technical colleges, schools, and universities for demonstrating the principles of a stroboscope without incurring the expense of a standard instrument.

NEW MORRIS VAN

AN entirely new Morris quarter-ton van, embodying many of the features of the Morris Minor, has been announced by the Nuffield Organization. It is equipped with the B.M.C. 800 c.c. overhead valve power unit and is to go into immediate production. The engine develops 30 b.h.p. at 4,800 r.p.m. and gives a maximum torque of 40 lb/ft at 2400 r.p.m. Independent suspension at the front is by long torsion bars with hydraulic piston-type dampers. At the rear, semi-elliptic heavy type springs are controlled by telescopic hydraulic dampers with anti-sway mounting. The van body has 78 cu ft capacity, 8 cu ft more than its predecessor. There is an additional space of 12 cu ft beside the driver. Large double rear doors give easy access; they have Triplex glass windows to allow good rear vision. Both export and home market models are going into immediate production.



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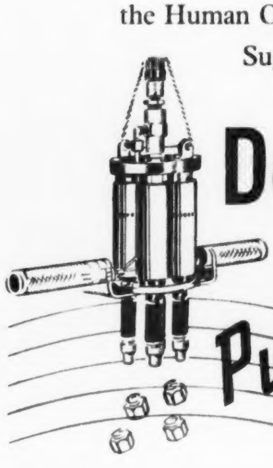

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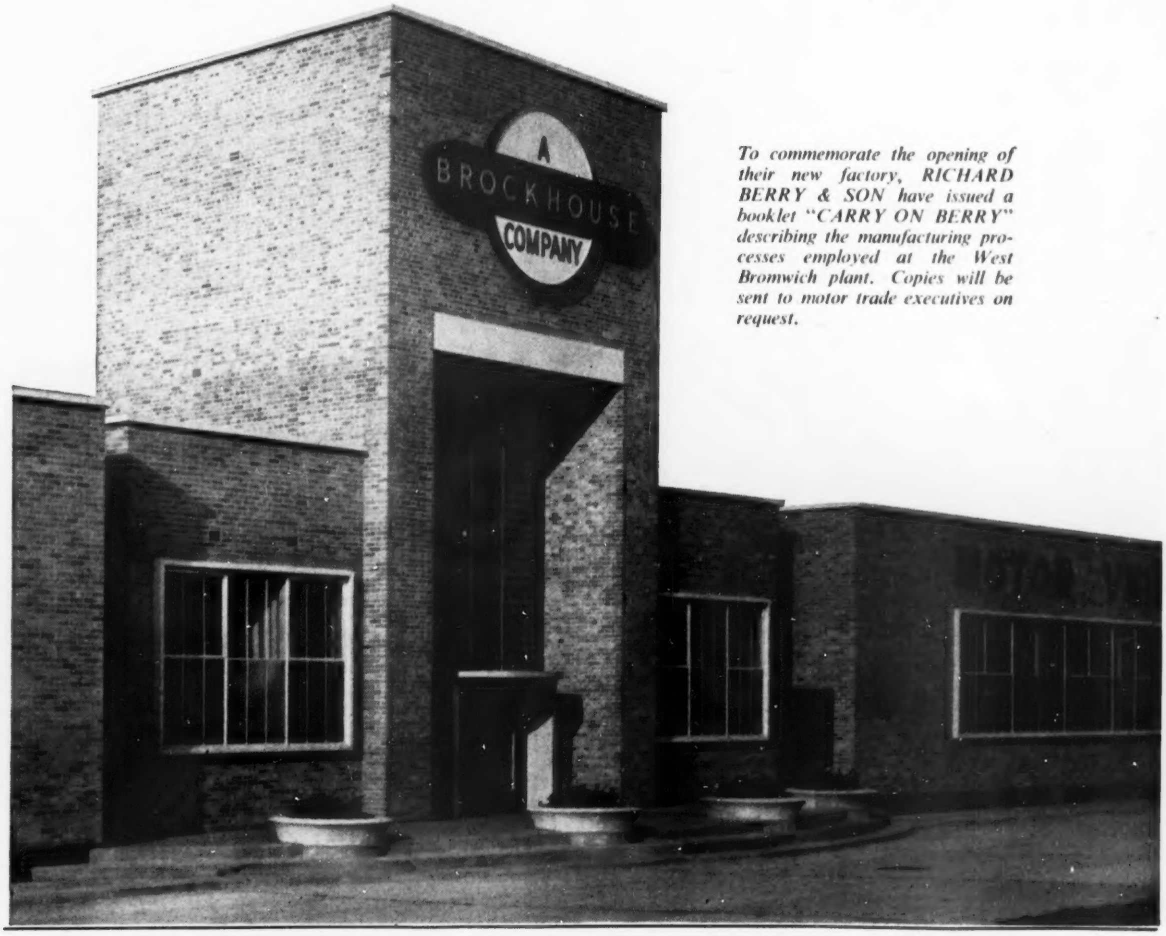

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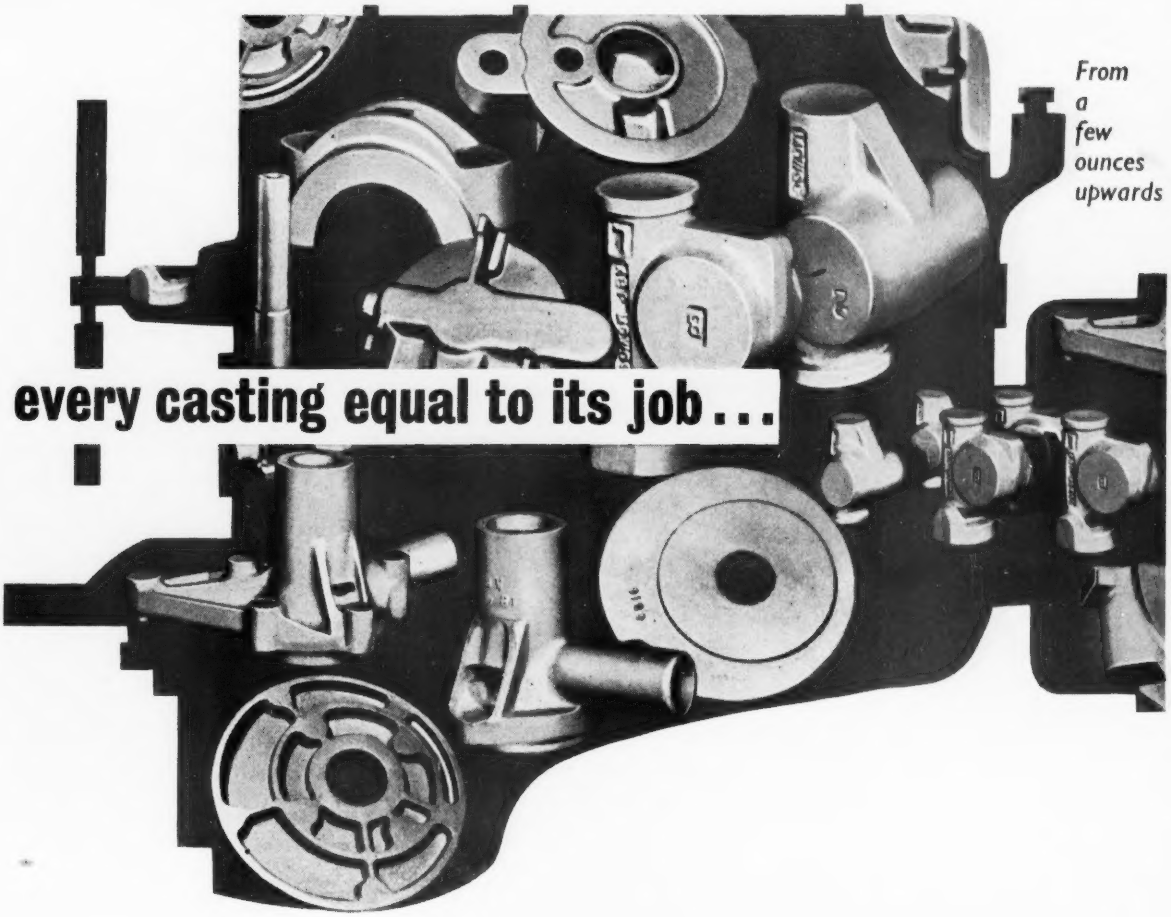

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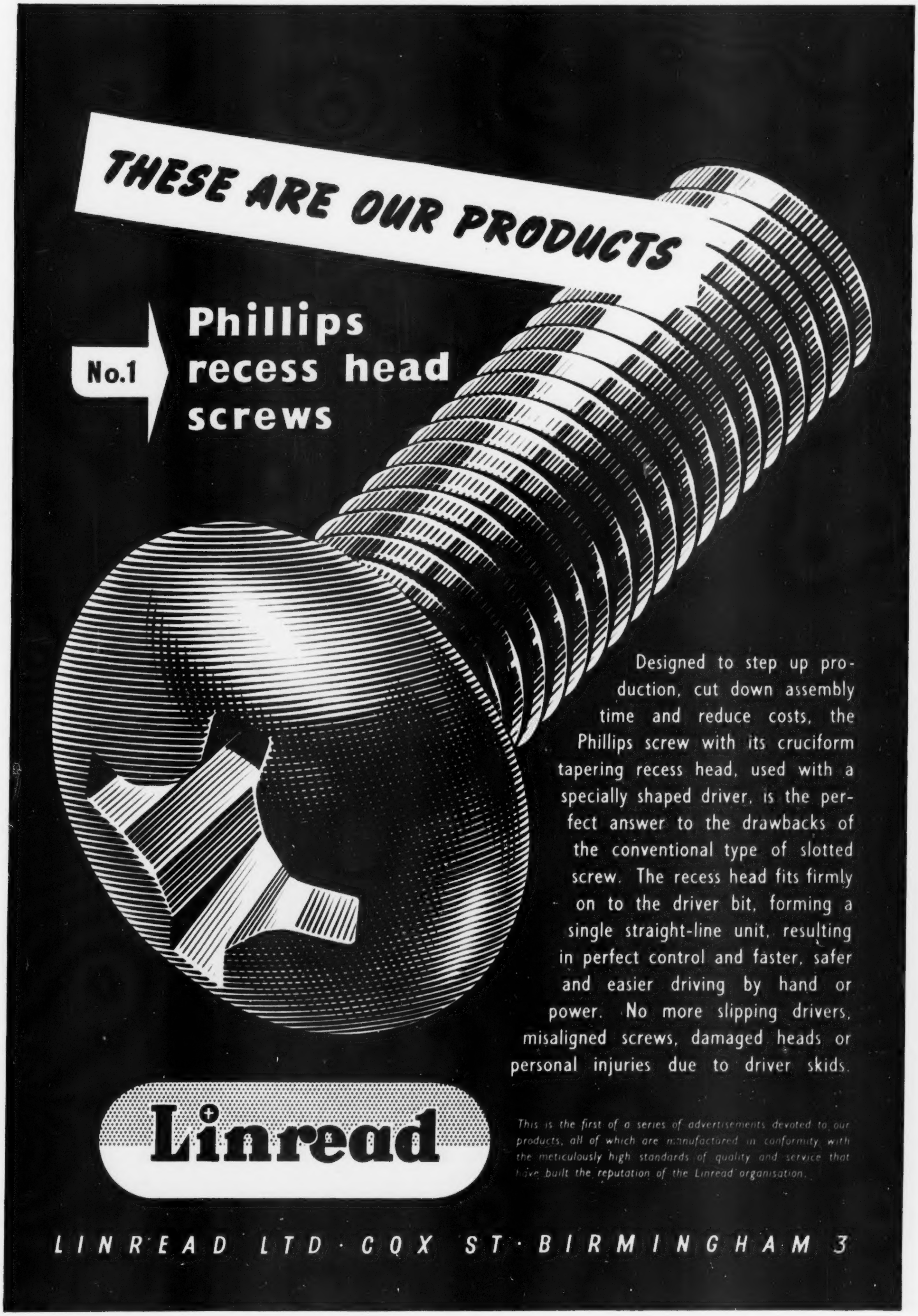
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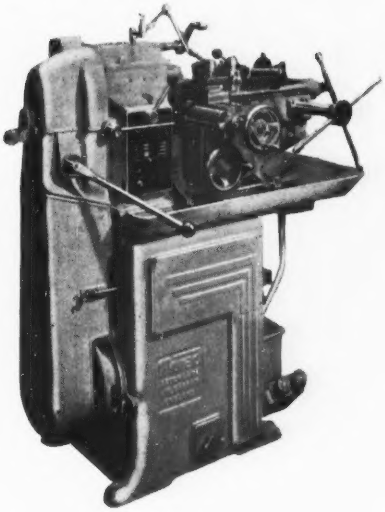

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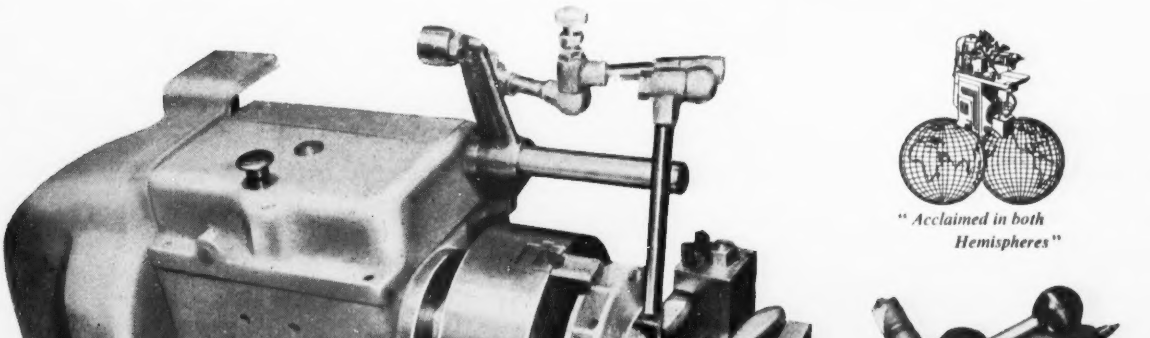

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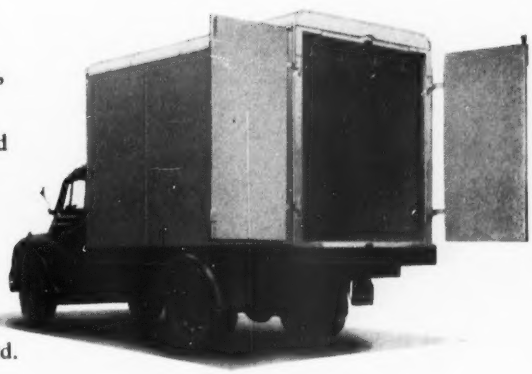

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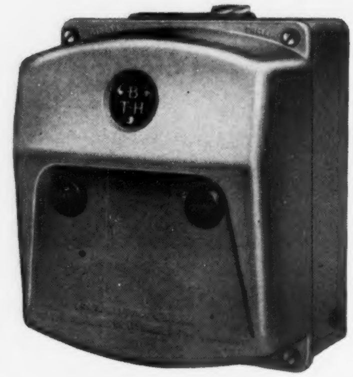

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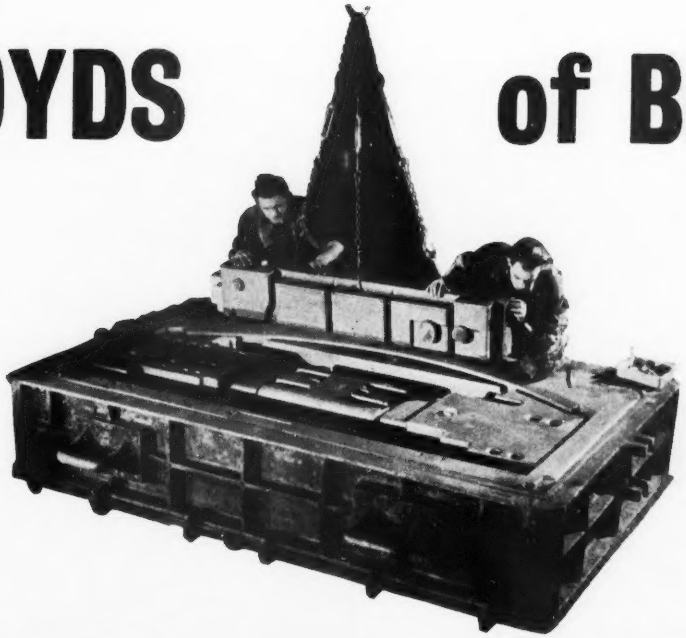

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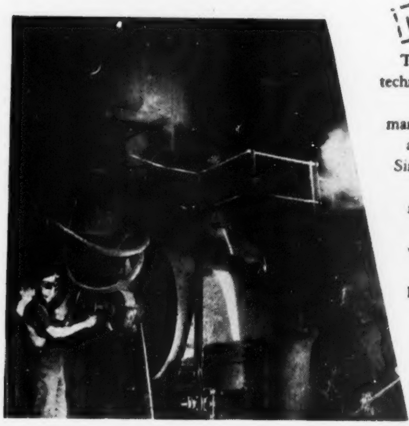

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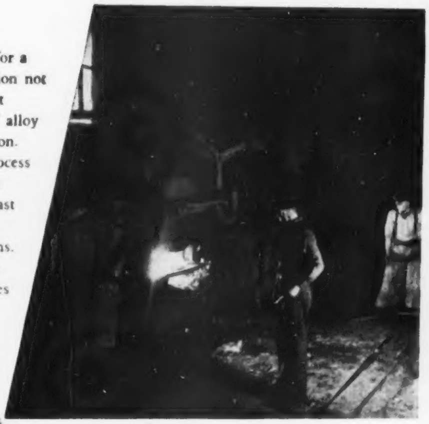

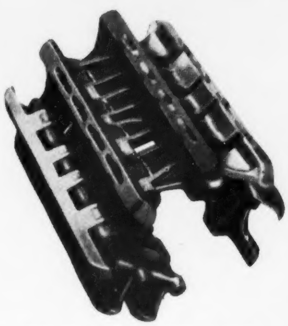

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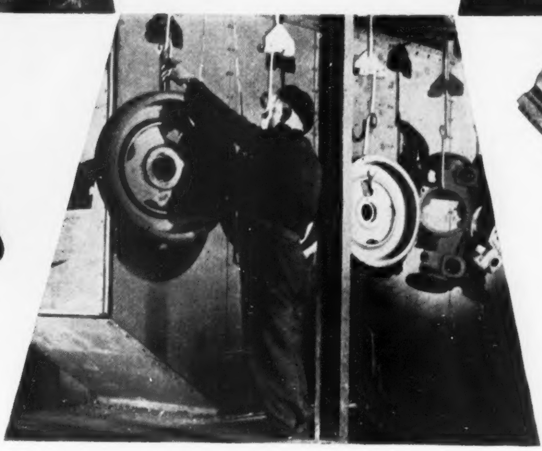

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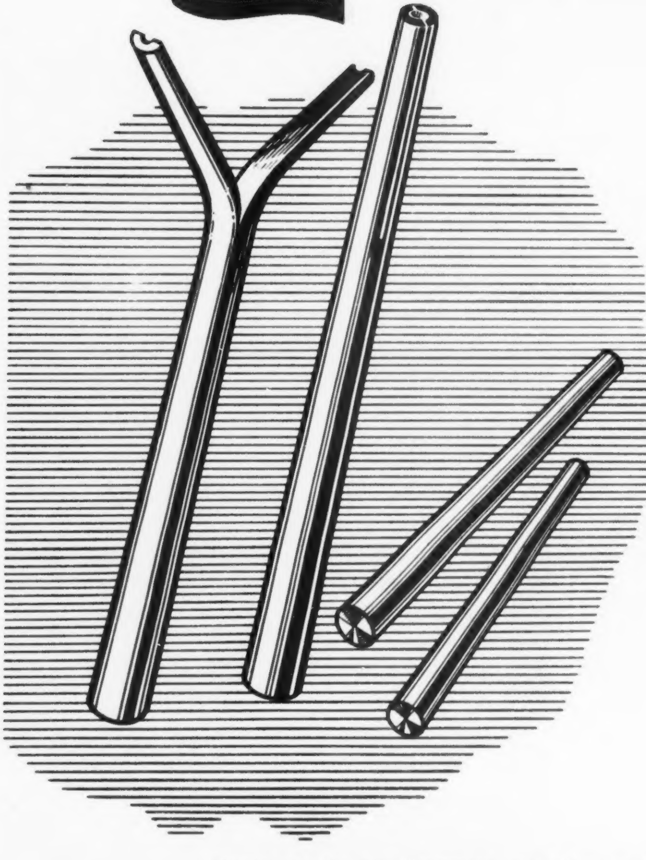

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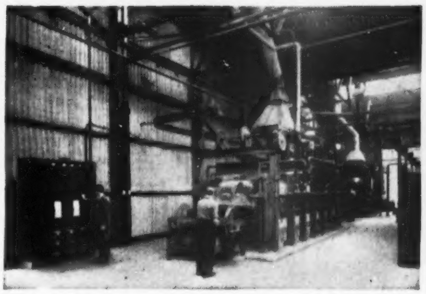

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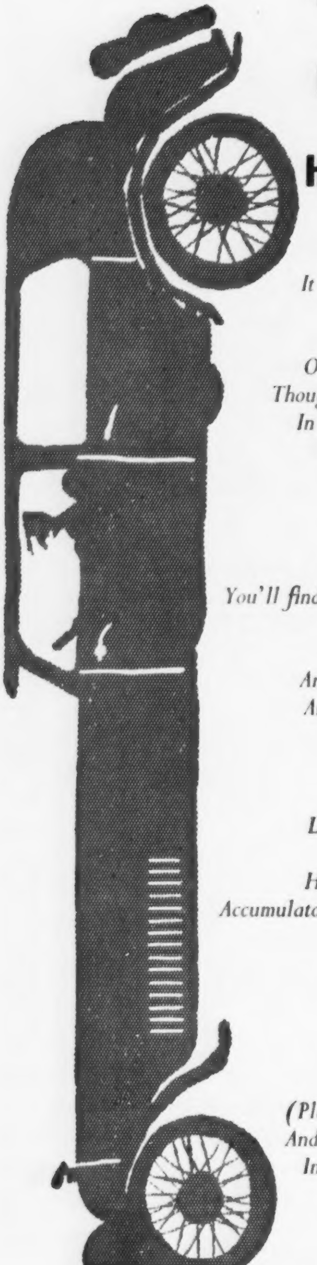
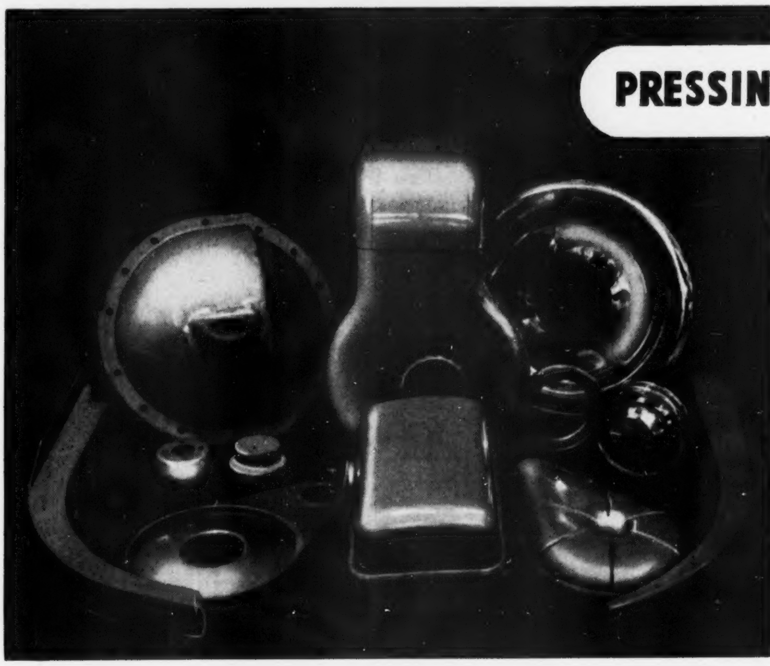


PRESSINGS & FABRICATIONS



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We manufacture our own Press Tools; Fabrications, Spinnings, Assemblies, and are able to undertake all kinds of Metal Finishes.

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RATE 4d. per word, minimum 4/-. Each paragraph charged separately. Box number 5 words—plus 1/-. Advertisements for the August, 1953, issue should be to hand not later than first post July 28th. No responsibility accepted for errors.

The engagement of persons answering these advertisements must be made through the local office of the Ministry of Labour and National Service etc. if the applicant is a man aged 18-64 or a woman aged 18-59 inclusive, unless he or she or the employer is excepted from the provisions of The Notification of Vacancies Order, 1952

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ARMSTRONG SIDDELEY MOTORS Car Division have a vacancy for a highly experienced Automobile Designer with considerable experience in the design of complete chassis, including engine and transmission.

THIS position offers interest and scope for a man of considerable experience and initiative. Reply fully in strictest confidence to Reference MC2, Armstrong Siddeley Motors, Coventry. [4677

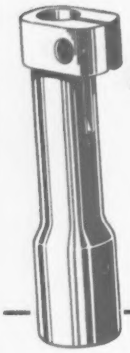
THE Villiers Engineering Company Limited, Marston Road, Wolverhampton, well-known manufacturers of two- and four-stroke internal combustion engines, require the services of an experienced Designer/Draughtsman for internal combustion engines and transmission work. A Degree or Higher National Certificate in Mechanical Engineering is desirable. The position offers unique opportunity and scope for a suitable man. A good salary with future prospects together with Staff Superannuation scheme is offered. Apply in confidence giving full particulars of education, experience, age and salary required to: Personnel Manager, at above address. [4686

DRAUGHTSMAN—Senior, with automobile experience, for work on new project.
HARRY FERGUSON RESEARCH LTD., Fletchamstead Highway, Coventry. [4687

TECHNICAL Assistants. Ford Motor Co. Ltd. of Dagenham, Essex, require two competent and progressive Engineers for Service Department. Both should have H.N.C. knowledge of I.C. engines, one should also possess expert diesel engine knowledge. Reply to Salaried Personnel Department, quoting reference R/TA. [4688

AUTOMOBILE Engineers. Engine and Chassis Designers and Draughtsmen, Styling Personnel, Body-Checker Draughtsmen required. See Ford Motor Company Ltd's displayed advertisement in this issue. [4689

SENIOR Designer required for Structural or Airframe Section of Guided Weapon Design Office. This man will be required to work on the Production Design of an accepted principle and applicants with both aircraft and/or motor industry experience will be considered most suitable. Ordinary or Higher National Certificate desirable. This post offers scope for advancement in an expanding organization. Address replies to Dept. C.P.S., The English Electric Co. Ltd., 336-7, Strand, W.C.2, quoting reference 845A. [4690



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GUY MOTORS Limited invite applications for the position of Chief Designer. Applicants must be well qualified and have wide experience of all classes of goods and passenger vehicle design for both home and export markets. Applications giving details of training, qualifications, experience, salary required, etc., to the Secretary, Guy Motors Ltd., Fallings Park, Wolverhampton. [4691

TECHNICAL Instructor for motor mechanics required by Kuwait Oil Company Ltd. for their Training Division in Kuwait. Applicants must have full recognized engineering apprenticeship (garage apprenticeship will not qualify) and at least 5 years' experience as a master tradesman. Preference will be given to candidates possessing an Ordinary National Certificate and having previous experience as an instructor. Age 32-40. Salary starting £790 p.a. clear, plus generous allowance, Pension Scheme and kit allowance. Write for application form giving brief details and quoting K.1727 to Box X/31, c/o 191, Gresham House, E.C.2. [4692

CATERPILLAR TRACTOR CO. Ltd., Coalville, Leicester, requires Chief Buyer, with experience of automotive industry, age 35/45. Applications, stating salary required and giving full information of experience and qualifications, will be treated in strict confidence and should be addressed to the Managing Director. [4693

A SENIOR Technical Assistant required for engine development work in Experimental Department of High Speed Diesel Manufacturer. Willingness to take responsibility, several years' experience of experimental work on C.I. engines, and at least H.N.C. standard are desired. Salary according to qualifications and experience. Application should state age, qualifications, and indication of salary required. Apply Personnel Manager, F. Perkins Ltd., Eastfield Factory, Peterborough, Northants. [4694

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PERMANENT position, good salary and pension scheme.

PLEASE apply in writing to the Secretarial Department, Commer Cars Limited, Luton. [4695

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WANTED—Old volumes of "Automobile Engineer," especially from 1941-1952.

PLEASE rush offers to: Mapleton House, 5412 16th Ave., Brooklyn 4, N.Y. (U.S.A.). [4696

PATENTS

THE proprietor of British Patent No. 577425, entitled "Improvements in Spring Suspension for Vehicles," offers same for licence or otherwise to ensure its practical working in Great Britain. Enquiries to Singer, Stern & Carlberg, Chrysler Building, New York 17, N.Y., U.S.A. [4683

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GERMAN factory fully experienced in production of high precision components for automobiles and engines, heat-treating and grinding a speciality, can quote orders at reasonable prices and short delivery. Has experience of English sub-contract work and fully equipped to deal with English threads and limits. Enquiries to Brauer, Schreyerstrasse 13, Kronberg i, Taunus, Germany. [4672

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Manufacturers of Plain Bearings in Gun Metal Steel and White Metal.
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The Personnel Manager, (Dept. A.U.), The De Havilland Aircraft Co. Ltd., Hatfield, Herts.

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